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QUESTION III:

The relationship between the vehicle and the track to ensure safety at high speeds.

A. Weight of vehicles per axle, position of the centre of gravity, wheel arrangement, layout to facilitate running through curves.

B. Track resistance, widening of gauge, radius of curves, superelevation, transition curves, points and crossings, check rails.

REPORT No. 3

(Germany, Denmark, Finland, Norway, Spain, Netherlands, Portugal and their Colonies, Sweden and Switzerland)

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Foreword.

As Question III deals with the vehicle and the track, two reporters were nominated in Germany, one of whom has specialised in rolling stock, and the other in the permanent way. The question has therefore been dealt with in two separate reports, linked together by a single questionnaire covering the two sections. The first part of this questionnaire, drawn up by the reporter on the part of the question relative to the vehicles, was also utilised with some additions by Mr. H. Deyl, Ministerial Councillor, the reporter of the Czechoslovakian State Railways, at Prague.

The part of the report dealing with the vehicles has been based upon the replies received to these questions. On the whole the report follows the division of matter adopted in the questionnaire but in drawing up the report this division has undergone some slight modifications.

Chapter I and II of the first part of the report, dealing with the *vehicles*, are principally of a discriptive and statistical nature. The reader will find therein a review of the information supplied to us and will learn to understand the more essential features of construction. Chapters III and IV contain considerations of the principles affecting the method of support and of guiding. The results and the conclusions are usually summed up immediately after the information given and are printed *in italics* so as to bring them out more noticeably; they do not therefore appear solely at the end of the different chapters. In various places *italics* are used to facilitate the search for the substance of the chapter and thereby to appreciate its contents, especially when the latter are of a descriptive character. Our task was to consider questions of principle rather than the statistical side of the subject. Some details were however unavoidable in chapters I and II, and also occasionally in chapter IV.

Replies were not received from all railways; some of the replies were received so late that we were not able to make use of the information contained therein in the statistical tables and in the text we could only take into account details of particular note.

The part of the report which deals with the permanent way also follows the division of matter of the questionnaire. This part of the questionnaire was drawn up in its main lines by Mr. Deyl, and completed in certain details by the author of the second part of the present report. The considerations developed therein are preceded by an investigation which deals with the relations between speed, safety and track. Here again, *italics* are used to facilitate the appreciation of the different questions dealt with and the results.

A. — THE VEHICLE.

CHAPTER I.

General arrangement of the vehicles; running gear.

1. — Steam locomotives and tenders.

For high speeds, locomotives with four-wheeled guiding bogies are used almost exclusively. Out of the 30 steam locomotives reported as suitable for high speeds and grouped by wheel arrangement, method of support, and country, in table 3 of Chapter II, 27 locomotives, i.e. 90 %, for maximum speeds of 80 to 120 km. (50 to 74.5 miles) an hour, have four-wheeled guiding bogies, and one locomotive the maximum designed speed of which is 100 km. (62 miles) per hour, has a guiding bogie consisting of a carrying pair of wheels and a pair of coupled wheels of the Krauss-Helmholtz type. Two locomotives only, the 2-8-0 and the 2-8-2 of the North of Spain Railways are fitted with a leading pony truck. The maximum speeds of these two locomotives have not been reported.

The wheel base of the bogies in generally 2.20 m. (7 ft. 2 5/8 in.) for 4-6-2 locomotives of standard gauge and the Spanish broad gauge; it is only 2.15 m. and 2.30 m. (7 ft. 5/8 in. and 7 ft. 6 9/16 in.) on two locomotives of this type. The differences are more marked in the case of the standard gauge 4-6-0 locomotives, with a bogie wheel base of 2 m. (6 ft. 6 3/4 in.), 2.15 m. (7 ft. 5/8 in.), 2.20 m. (7 ft. 2 5/8 in.) and 2.30 m. (7 ft. 6 9/16 in.). In the case of the 4-6-0 broad gauge locomotives of the Finnish [1.524 m. (5 ft.)] and the Spanish [1.67 m. (5 ft. 5 3/4 in.)] railways, the bogie wheel base is 2 m. (6 ft. 6 3/4 in.). The 4-6-2 locomotives of the Dutch East Indies State Railways, of 1.067 m. (3 ft. 6 in.) gauge, have bogie wheel bases of 1.80 and 1.95 m. (5 ft. 11 in. and 6 ft. 4 25/32 in.); in the case of the 4-6-4 locomotives of the same railway the wheel base is 2 m. (6 ft. 6 3/4 in.).

The *rigid wheel base* of steam locomotives, i.e. the wheel base of the outer coupled pairs of wheels rigidly mounted in a common frame, varies as would be expected, widely; it depends among other things, on the number of coupled wheels (22 locomotives have 3 pairs of coupled wheels and 8 locomotives have 4), and upon the diameter of the coupled wheels; it differs, moreover, according as the brake blocks are placed on a line through the horizontal centre line of the wheels as on the standard German State Railway locomotives, or lower down, and according to the base or length of support considered appropriate to the length of the vehicle. Among other factors which contribute to good riding and which reduce the striking angles during hunting movements on the straight, i.e. at high speed, the rigid wheel base must be first mentioned. It is a good thing to make it as large as possible: the turnouts and curves govern the limits. The rigid wheel base in former times was often restricted by regulations which specified that it should not exceed a certain numerical value. This procedure was materially wrong, as above all the arrangement of the vehicle must be such as to assure the greatest possible safety in running. The rigid wheel base therefore should always be determined solely after consideration of the whole of the running gear of the vehicle and of the parts connecting these details together; care should be taken that the running gear as a whole can without difficulty pass through the turnouts and curves without any wedging action.

An obvious condition when the guiding is to give the required degree of safety is that geometrically the running should be irreproachable. A locomotive which with its tender can easily pass over points and crossings will not experience any difficulty in running through curves on the running line. Frequently the angles of intersection of the old types of turnouts had the result of making the maximum values of the striking angles on curves on running roads double their own value. We have therefore quite naturally decided not to quote the dimensions of rigid wheel bases taken from the drawings sent us.

One of the principal factors as regards the guiding pressure between the guiding wheel of the first pair of wheels and the rail, i.e. as regards the relationship between this transverse force and the wheel load, is *the guided length of the vehicle*, or in other terms the distance separating the parts of the vehicle (pairs of wheels, bogie centres) which govern the position of the horizontal axis of the vehicle relatively to the track. The guiding force acting on the wheel which comes into contact with the rail on a curve has to overcome all the slipping resistances at each wheel at the same time supposing that no other wheels are hard against the rail nor take any part in the guiding (cf. Chapter IV); this makes the vehicle turn about the « friction centre » with a leverage equal to its distance from the

perpendicular through the centre of gravity and the longitudinal axis of the vehicle. This distance differs from the guided length. In a general way the leverage increases with the increase of the guiding length. The greater this length proportionally to that of the rigid wheel base, the smaller the turning force acting on the straight and on the curves on the longitudinal axis of the vehicle, and the smaller the transverse pressure (guiding pressure) of the flange of the guiding wheel pressed against the side of the rail, and with it the risk of the guiding wheel climbing the rail. Any tendency of the guiding wheel to mount the rail must therefore be prevented by all means possible. The first of these means consists in reducing the guiding pressure as much as possible and making certain that there is sufficient weight on the leading wheel. These questions are considered more closely in Chapters III and IV of the present report. All that need now be said here is that, as regards the numerical value of the ratio of the guided length to the rigid wheel base, that it varies between 1.856 and 1.510 for the high speed steam locomotives mentioned in the replies. The maximum value 1.856 is that of the 4-6-4, series 28, locomotive of the Dutch East Indies State Railways [1.067 m. (3 ft. 6 in.) gauge]; then the 4-6-0 Netherlands and Norwegian State Railways locomotives with 1.833; the Danish State 4-6-0 with 1.815; a former Saxon 4-6-0 locomotive with 1.797; the other German locomotives have values of 1.713 to 1.63, this latter value being that of the standard 4-6-2, series 01, locomotives. The smallest values are those of the 4-8-0 and 2-8-0 types, broad gauge, Spanish locomotives which have up to 5.70 m. (18 fr. 8 7/16 in.) rigid wheel base.

A question of principle is whether, in addition to the directing wheels of the bogie, the first pair of wheels of the rigid wheel base ought to take part in the guiding on small radius curves. At its 1909 Convention, the German Railway Association made the following recommendation on this point:

> It is recommended that the bogie be given sufficient lateral play for the flanges of the first pair of wheels rigidly mounted in the frame to come into contact with the outer rail on the sharpest curves on the running lines.

This point of view was abandoned in the new edition of 1930. This change of opinion will be understood if it is realised that the guiding is chiefly compromised by the transverse forces set up in the form of shocks, and which may always occur, seeing that irregularities in the lining of the track are inevitable. These phenomena have been carefully enquired into and the important part played by the mass acting at the point of shock, known as « reduced mass » which is notably smaller at the bogies wheels placed at a considerable distance ahead than at the first coupled pair of wheels of the rigid wheel base, which is much closer to the centre of gravity of the locomotive, in these transverse forces seen as shocks, has been recognised. The value of the side play of the bogie and of the details of the running gear which cause the locomotive, when running through a curve, to take such a geometrical position that the guiding can be assured by a single point or by two points of the vehicle depends upon the solution given to this question of principle. Some 12 administrations expressed their opinion on this matter: 7 state that they prefer guiding by the bogie centre alone, and 5 consider that the first pair of rigid wheels in the frame should also play some part in guiding, although of these 5 companies 2 add that on curves of less than 300 m. (15 chains) radius and on turnouts the guiding should be assured by the bogie alone. This shows that there is a certain diversity of detail in the running gear.

It would be useless however to give

at this point particulars of the wheels with side play and those with thin flanges, as the conditions of the turnouts which the locomotives have to run through are not known. Nonetheless investigations have been made upon a number of steam locomotives on a curve of 300 m. (15 chains) radius and in order to be able to compare them upon the same basis, on the 1 in 8 Baden turnout [r=165.41 m. (8 1/4 chains)] and the new German State Railways' 1 in 6.6 turnout [r=190 m. (9 1/2 chains)]. For this enquiry the following engines were selected :

The 2-8-2, series 19, of the German State Railways, with Krauss-Helmholtz bogie and a rigid wheel base of 4 m. (13 ft. 1 1/2 in);

The 4-8-2 of the Madrid-Saragossa-Alicante Railways with a rigid wheel base of 3.70 m. (12 ft. 1 43/64 in.);

The 4-6-4 tank locomotive of the Netherlands Railways with a 4.20 m. (13 ft. 9 23/64 in.) rigid wheel base; and

The 4-8-4 tank locomotive of the North of Spain Railways with 5.10 m. (16 ft. 9 in.) rigid wheel base and only 45 mm. (1 49/64 in.) bogie side play, on each side of the centre.

The result of this investigation is that on a curve of 300 m. (15 chains) radius the leading pair of coupled wheels of all these locomotives takes its part in guiding the locomotive. This is even more the case on curves of greater radius and also momentarily on the straight, should the vehicle hunt. In the turnouts considered, the guiding is due almost exclusively to the bogie. An exceptional case is that of the 4-8-2 locomotives of the Madrid-Saragossa-Alicante Railways, on which, when passing through the 1 in 6.6 turnout of the German State Railways, the first pair of coupled wheels continuously takes part in the guiding.

We have mentioned these detailed examples in this Chapter I to show to what extent the side play, the flange thickness, etc., ought to vary with the design of the turnouts. This is why we have decided not to deal more fully with this matter at this point, especially as the question of *guiding* is considered as a whole in Chapter IV.

The tenders do not give rise to any special remarks.

2. — Electric locomotives.

Among the 18 electric locomotives more or less explicitly dealt with in the replies to the questionnaire, 4 locomotives have not been considered in the present report, because their speed is only 65 and 70 km. (40.4 to 43.5 miles) per hour or because, owing to the small diameter of their driving wheels and the arrangement thereof, they do not appear to be suitable for high speeds. *Table 4 of Chapter II gives a general review of the wheel arrangement, methods of support and the owning administrations of the 14 locomotives still to be examined in this report.*

The maximum speeds of the German State Railway locomotives are from 90 to 110 km. (56 to 68.3 miles) per hour; the electric locomotives of the Swedish State and the Dutch East Indies State Railways can run up to 90 km. (56 miles) per hour. The North of Spain Railways have not given the maximum speeds of their electric locomotives.

The high-speed electric locomotives are fitted with many different types of control and driving gear. Some of the locomotives have one or several motors carried above the frame, the rotary motion thereof being transmitted by inclined rods to one or two quills and from these by coupling rods to the driving wheels (the 4-6-4, series E 06, and the 4-8-2, series E 5, of the German State Railways), while the most recent locomotives without exception have individual axle drive on the Buchli, Westinghouse, and Linke-Hoffmann systems or are driven by axle-hung (nose-suspended) motors. These locomotives are fit-

ted with as many as 8 traction motors.

The electric locomotive, like the steam tank engine, has to run equally well in both directions. It might be thought in consequence that four-wheeled bogies consisting of either two pairs of carrying or one pair of carrying and one pair of driving wheels would be used in about the same proportion as on steam locomotives. This is not the case however, as *only 64.3 % of the electric locomotives have four-wheeled bogies* as compared with 93 % of the steam locomotives. The reason for this is that the electric locomotives do not as a rule have to be so heavy per horse power as the steam locomotives. For example, the average weight per electric horse power (continuous rating) of the 1 Do 1 locomotives of the E 16, E 16[5] and E 17 series of the German State Railways is 38.3 kgr. (83.9 lb.), instead of 68.2 kgr. (150 lb.) for the 2-8-2 series 39°-2 (old P 10 class) steam locomotives of the same railway. The required continuous output can be obtained in the first case by a lighter construction than in the second and consequently fewer pairs of wheels are needed.

The result of this is that on 28.6 % of the electric locomotives designed to run in both directions and on a further 21.4 % in one direction, a two-wheel pony truck (Bissel bogie) is used to guide the engine at the leading end, whereas in the case of steam locomotives the proportion is only 7 %. Motor bogies, not used on high-speed steam locomotives, are used at the leading end of 7.1 % of the electric locomotives.

The *wheel base of the four-wheeled carrying bogies* is 1.65 m. (5 ft. 5 in.) on the old E 50 type of the German State Railways, and 1.80 to 2 m. (5 ft. 11 in. to 6 ft. 6 3/4 in.) on the newest locomotives; it is 2.20 m. (7 ft. 2 5/8 in.) on the 2 Co-Co 2 North of Spain broad-gauge locomotives. In the case of bogies consisting of one pair of carrying and one pair of coupled wheels, the German State Railways use a wheel base of 2.70 m. (8 ft. 10 5/16 in.) with the Krauss-Helmholtz type, and the Dutch East Indies State Railways [1.067 m. (3 ft. 6 in.) gauge,] 2 m. (6 ft. 6 3/4 in.) with the Buchli (Java bogie) type. The constructional details of these bogies are given in Chapter II and of those the pony trucks in chapter IV.

The reasons determining the rigid wheel base were briefly given in paragraph 1 above (steam locomotives). It is consequently not surprising that the German State Railways should have made the rigid wheel base of the 1 Do 1, series E 16[5], engines with a leading pony truck the exceptional length of 6.10 m. (20 feet) when it is also realised that this locomotive has been built for a maximum speed of 110 km. (68.3 miles) per hour, and has a pony truck at both ends. Such long rigid wheel bases are not found on steam locomotives ; the maximum of 5.70 m. (18 ft. 8 7/16 in.) is that of the 4-8-0 locomotive of the North of Spain Railways. On the other hand there are electric locomotives without rigid wheel base. In the saturated steam era, the 2-4-4 locomotives of the former Bavarian State Railways, which were remarkable for their speed, also had no rigid wheel base. To return to the electric locomotives, the 1 Do 1, E 16 series, for example, of the German State Railways, has no rigid wheel base. It has at both ends Krauss-Helmholtz type bogies, while the two middle pairs of wheels have a certain amount of side play. To a large extent the main frame of the locomotive is only guided by the centre pins of these bogies. The conditions are like those of the 1 Do 1, E 17 series, of the German State Railways, on which the two middle pairs of wheels have also a certain amount of side play which has been further increased by thinning the flanges of the tyres on these wheels. The 2 D 1, E 50 series. and the 2 B B 2, E 52 series, belonging to the same railway, have only one pair of

wheels rigidly mounted in the frame, so that here again there is no rigid wheel base.

The guided length of the electric locomotives cannot be given accurately for all the locomotives, especially when the guiding pairs of wheels are separate pairs of carrying wheels. The Administrations supplying information on this subject take the guided length as the distance between the carrying pairs of wheels and the last pair of wheels rigidly mounted in the frame, or to the centre pin of the trailing bogie. If the leading bogie is not fitted with self-centring gear, it can only exert any guiding action on the main frame when its full side play has been taken up. This occurs on sharp curves, but can also take place in the straight through hunting. Apart from this the only forces in play are the relatively small frictional resistances acting in the manner of a guiding force on the longitudinal centre line of the vehicle; they do not have the same effect as a proper self-centring gear, especially as regards the inevitable transverse shocks. These carrying pairs of wheels cannot ordinarily be considered as « carrying directing wheels ». In his note on the « Effect of masses under sudden changes of direction on the running of railway vehicles » (*Organ für die Fortschritte des Eisenbahnwesens,* 1931, p. 271) Dr.-Ing. Uebelacker has clearly shown the great importance of spring controlled self-centring gear or, in other words, of spring controlled guiding. The directing force is absorbed by the elastic yielding of the locomotive frame, of the carrying springs, and lastly of the rails, if no special self-centring gear is provided for the purpose. *This shows how superior the four-wheeled bogie fitted with the modern design of centring gear including a pair of laminated springs is to the two-wheeled truck from both the static and the dynamic points of view.* If this has been appreciated, the difficulties in forming a clear and precise conception of what is meant by the guided length will be recognised, when it is a question of the guiding being obtained with the help of a pony truck which, if it is to direct naturally, needs suitable guiding springs. If in addition this pony truck is followed by a number of pairs of coupled wheels with side play and possibly thin flanges, it becomes necessary to decide in each case how by the law of equilibrium between the centring force and the frictional resistances the statical inscription of the vehicle actually takes place. These difficulties have made impossible any numerical comparison on the subject of the « guided length » of electric locomotives like that given above for the steam locomotives.

We have already explained that the question of the guided length is closely connected with that of knowing if the first pair of wheels rigidly mounted in the frame should or should not take part in guiding the locomotive. The German State Railways alone answered this question: their reply being as follows:

The guiding must always be controlled by the centre pin of the leading four-wheeled bogie alone. In the case of a pony truck, the first pair of driving coupled wheels should also be made to play its part in the guiding of the engine, provided it is not more than 2.8 to 3 m. (9 ft. 2 15/64 in. to 9 ft. 10 1/8 in.) from the carrying pair.

On the electric locomotives of the German State Railways the position is as follows:

On the 1 Do 1, series E 16[5] locomotives, the pony trucks of which have 110 mm. (4 11/32 inches) side play, the leading pair of wheels is 3.40 m. (11 ft. 1 7/8 in.) from the leading coupled wheels, and the rigid wheel base is 6.10 m. (20 feet), the truck alone should guide. For this purpose, it has been fitted with centring gear, the initial tension on the spring being 870 kgr. (1 918 lb.) and the final, 2750 kgr. (6062 lb.). In the case of the 1 Bo-Bo 1, E 18 series, each pony

truck of which has 80 mm. (3 5/32 inches) side play, the carrying pairs of wheels are 2.30 m. (7 ft. 6 9/16 in.), from the leading driving and the rigid wheel base is 3.40 m. (11 ft. 1 7/8 in.) the first pair of driving wheels should also play its part in the guiding. The tension of the centring gear is from 1755 to 2590 kgr. (3869 lb. to 5709 lb.). The 2 Do 1, E 21[01] series, locomotives have been built in two classes E 21[01] and E 21[02]. The E 21[01] has a pony truck the wheels of which are 2.05 m. (6 ft. 8 45/64 in.) from the first rigid pair of wheels. The two middle pairs of wheels have a certain amount of side play as well as thin flanges; the 4th pair of wheels has 5 mm. (13/64 inch) side play. This pair can also be considered as being rigid as the play does not exceed the allowable amount of wear of the axle collars. This locomotive has therefore a rigid wheel base of 6.60 m. (21 ft. 8 5/32 in.). The second type, the E 21[02] has a Krauss-Helmholtz bogie with a 2.65 m. (8 ft. 8 23/64 in.) wheel base. The two middle pairs of driving wheels have thin tyres and can move transversely. The 4th pair of driving wheels is rigid and is in fact the only rigid pair, so that here again the rigid wheel base may be considered as *nil;* the guiding is, however, assured not by a bogie and a pony truck, but by two four-wheeled bogies. The 2 D 1, E 50 series, locomotive has one rigid pair of wheels; all the other pairs of driving wheels can move transversely. The guiding is assured in one direction of running by the pony truck. The maximum speed of this locomotive is only 90 km. (56 miles) per hour, however. The electric locomotives of the German State Railways for maximum speeds of 110 km. (68.3 miles) an hour are only designed so that there is no rigid wheel base when they are guided by four-wheeled directing bogies and not by pony trucks.

As with steam locomotives, the question of the side play of individual pairs of wheels and of bogies on a curve of 300 m. (15 chains) radius with 32 mm. (1 17/64 inches) play between the flanges and the rails on the German State Railways 1 in 6.6 [r = 190 m. (9 1/2 chains)] turnout, has been investigated. The locomotives considered had at least six pairs of wheels.

The result of this investigation is briefly as follows:

The most important position on the turnouts is that in which the first pair of coupled wheels coming on to the crossing is in line with the intersection of the curve of the turnout and of the straight line. With approximately the normal guiding action, not a single pair of wheels of any of these locomotives takes part in the guiding. On the other hand under what is called dynamic inscription, i.e. under the influence of the centrifugal force all the driving wheels may come into contact with the outer line of rails except only in the case of the 1 Do 1, E 16 series. For the reasons given at the end of paragraph I-1, it would be of little use to go into further details here.

To sum up, the following may be said as regards I-1 and I-2:

The whole of the running gear of the steam locomotives, the tenders and the electric locomotives as well as the coupling together of the different parts of the vehicle should be so arranged that turnouts and curves can be run through without restriction.

In agreement with most of the Administrations we recommend that the widest possible use be made of four-wheeled bogies for guiding, that the longitudinal centre line of the vehicles should be controlled by the leading bogie centre alone, and that the guided length be the greatest possible. When single pairs of carrying wheels are used for guiding, the use of relatively strong self-centring gear is presupposed. The running gear

of electric locomotives and steam tank engines should be designed so that they run equally well in both directions and these locomotives should have a sufficiently long rigid wheel base for it to come into play when the guiding by an individual pair of wheels is not altogether satisfactory.

The electric locomotives of the German State Railways fitted with guiding pony trucks and in which the pairs of wheels in the main frame only take part in the guiding in the event of considerable lateral movements, run quite satisfactorily up to the maximum speed of 110 km. (68.3 miles) per hour, because their frames (main frame or also motor bogies) have a sufficiently long rigid wheel base relatively to the distance of the carrying pair of whels. Through the guiding action of the wheel flanges, and on the straight of the friction due to adhesion, this rigid wheel base forms the constant base from which the pony truck assisted by a centring gear of sufficient capacity transmits the directing force to the track without transverse shocks or oscillations. In the case of locomotives with motor bogies (the rigid wheel base of which varies from less than 4.00 m. to 4.50 m. (13 ft. 1 1/2 in. to 14 ft. 9 5/32 in.), we consider the bogies should be coupled together to prevent the free rotary oscillations of each bogie taken separately about the vertical axis of rotation, so that at high speeds on the straight all the pairs of wheels rigidly mounted on the two bogies would take up much the same working position as if they were carried in one continuous rigid frame.

All the electric locomotives for fast trains with the wheels carried in the frame with transverse play relatively to the track have been built by the German State Railways with four-wheeled guiding bogies and have given good results, as regards running at speeds up to 110 km. (68.3 miles) per hour.

3. — Carriages.

As regards carriages for high speeds, bogie carriages especially and some four-wheeled carriages have been reported. With very few exceptions we shall only consider modern stock in this report. A few old types built in some instances 29 years ago are dealt with for the sake of comparison, or because the owning company has not mentioned any newer vehicles. In all 50 types of carriages will be considered, namely 45 bogie carriages including 5 electric rail motor coaches (cf. Table 1); 4 four-wheeled carriages (1 Swedish State, 2 Bergslagen Railway, 1 North of Spain Railways) and one articulated unit composed of 4 bodies on 3 Jakobs bogies and two radial pairs of wheels (Swedish State).

In this question with which we are dealing, the running gear and the method of support are again important. *We will begin by considering bogie carriages.*

The reason why bogie carriages are used in so great majority is well known. We can only build long vehicles without excessive overhang, and without too great a striking angle, and in conjunction with multiple springing get really smooth running if we use bogie carriages. When multiple suspension is used, the bogie can absorb the greater part of the vertical and horizontal shocks, and hunting movements, and the body is thereby isolated from the effects thereof.

All the bogies are fitted with bolsters carrying the bodies of the vehicles. Bogies without bolsters, in which the body is carried directly on the bogie frame, were long ago recognised as unsuitable for carriages for express trains (cf. among other articles the *Organ* 1912, 14th supplementary volume, p. 151), and are now rarely used under vehicles of this kind. *The bolsters are carried either on transverse springs* — today this is practically the only arrangement in use — *or on longitudinal springs.*

The bogies are fitted with either

double or triple springing or suspension; the frames are built up of steel pressings, rolled sections, or cast steel.

Table 1 below gives particulars of the types of bogie carriages of the different railways, classified according to these different features.

With the assistance of this table, we will rapidly examine the different methods of construction.

None of the bogies fitted with triple suspension, with bolster springs arranged transversely, is fitted with longitudinal equalising levers, so that the bogies are supported each in two transverse planes on four points.

Consequently the bogies with triple suspension with bolster springs arranged transversely are like one another from the point of view of the method of suspension and of the method of carrying the load (bearings).

The weight of the body is carried through a bolster centre and two side friction blocks, by the body bolster. This bolster is carried in the well-known way, though double laminated springs arranged transversely, by the bottom bolster which in its turn is supported by means of 4 inclined links from the cross bearers of the bogie frame. Bolsters with vertical links are not used on these bogies. The bogie frame is carried by four laminated springs resting on the axle boxes, the spring hangers being fitted with auxiliary helical or volute springs. Most of the administrations have adopted pressed steel for the bogie sole bars (14 types of bogies); some railways prefer bogie frames built up of rolled sections (5 types).

For some years the bogies with triple suspension of some railways have had the bearing springs arranged longitudinally. This method of construction makes it possible to use longer bolster springs with a particularly large deflection and makes them more accessible (for setting up to original height, brake blocking in service). As regards the method of support, the same arrangement is used as in the cases examined previously. The German State Railways have designed bogies of this type for passenger carriages, based on the types II and III Görlitz bogies, the heavy type II Görlitz

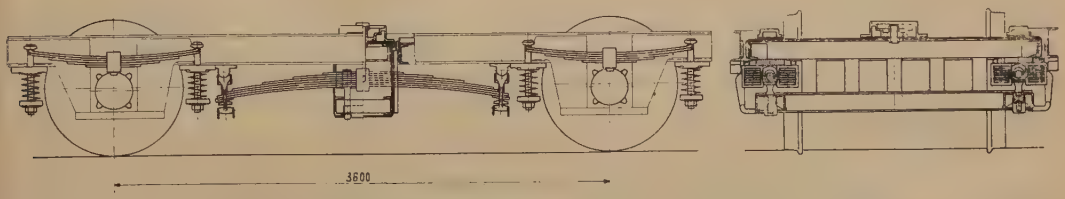


Fig. 1. — Görlitz type II (heavy) bogie, German State Railways.

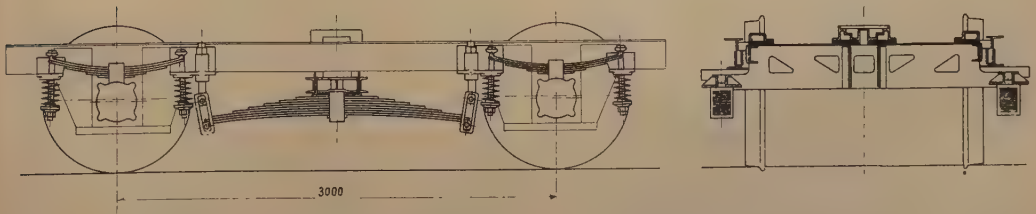


Fig. 2. — Görlitz type III (light) bogie, German State Railways.

TABLE 1.

Among the 45 bogie carriages, including 5 motor coaches, there are:

1	2	3	4	5	6
Units.	With	Bolster springs arranged transversely and frames of			With longitudinal bolster springs and frame of steel rolled sections.
		Steel pressings.	Rolled sections.	Cast steel.	
26	Triple suspension.	3 DR (*Pr*, *Wü*, *Sa*).	5 DR (*Bav*, *Bad*).		3 DR (Görlitz design).
		1 DR (6-wheeled).			1 S (Görlitz design).
		4 *Fin*.			3 NI.
		1 *Nor*.			
		1 S.			
		3 MZA.			
		1 *SpN*.			
	Total.	14	5		7
16	Double suspension.	5 DR (*Pr*, *Wü*, *Sa*).	3 NS (Equalisers under axle boxes).	2 *SpN*.	
		1 DR (6-wheeled).			
		2 NS.	2 *Dan*,		
			1 BJG (Equalisers above axle boxes).		
	Total.	8	6	2	
42		22	11	2	7
+ 3	(3 NI, no detailed information).				
45					

Explanation:

DR = German State Railway Company.
Pr = Prussia (former State Railways).
Wü = Wurtemberg (former State Railways).
Sa = Saxony (former State Railways).
Bay = Bavaria (former State Railways).
Bad = Baden (former State Railways).
Dän = Danish State Railways.
Fin = Finnish State Railways.
NI = Dutch Indies (State Railways).
NS = Netherlands Railways.
N = Norwegian State Railways.
S = Swedish State Railways.
BJC = Bergslagen Railway (Sweden)
SpN = North of Spain Railways.
MZA = Madrid-Saragossa-Alicante Railways.

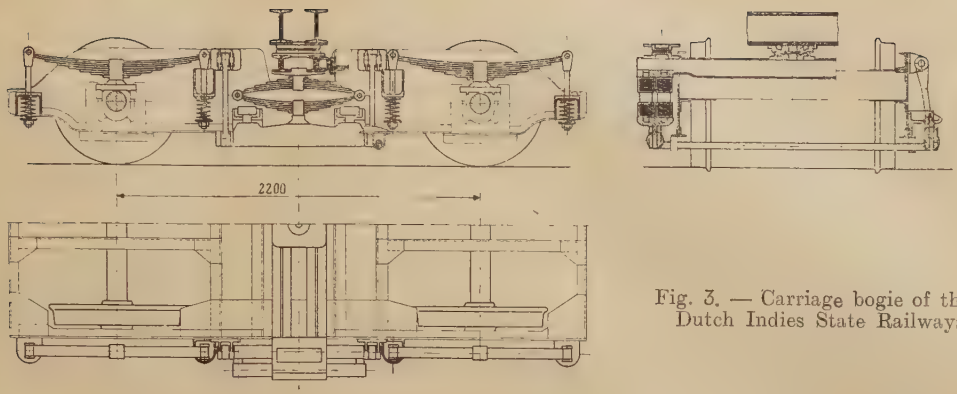


Fig. 3. — Carriage bogie of the Dutch Indies State Railways.

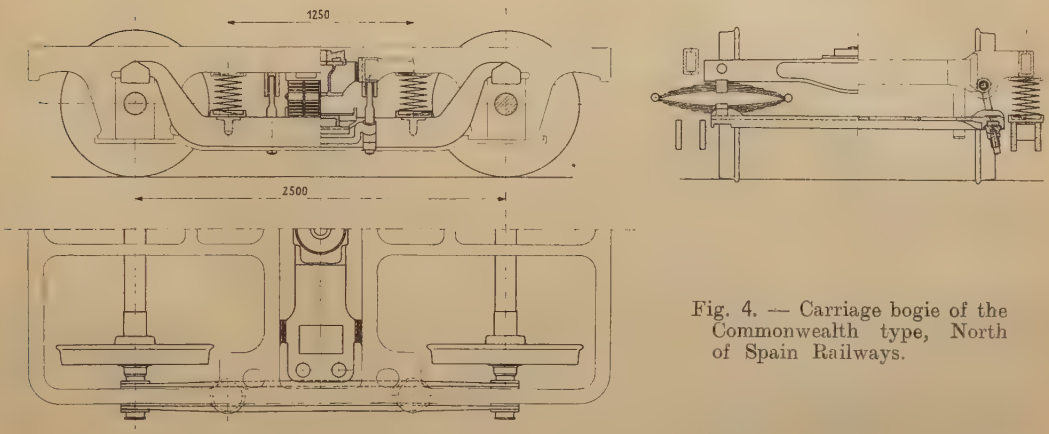


Fig. 4. — Carriage bogie of the Commonwealth type, North of Spain Railways.

with 3.60 m. (11 ft. 9 3/4 in.) wheel base being used under coaches for express trains, and the light type III Görlitz with 3 m. (9 ft. 10 1/8 in.) wheel base under coaches for ordinary and stopping trains. In both cases the frame is built up of rolled sections and three sets of springs are used.

In the case of the heavy type II Görlitz bogie (fig. 1), the load on the body bolster is transmitted by two swing links free to move transversely to a bolster and therefrom to two longitudinal bolster springs arranged side by side. The ends of these longitudinal springs are shackled to the bogie frame.

The light type II Görlitz bogie (fig. 2) is not fitted with special swing links; in place of these the longitudinal springs of the bolsters are carried by shackles so as to be able to move transversely an amount equal to the transverse play of the bolster. The bolster in this case is supported directly by the buckles of the bolster springs.

The Dutch East Indies State Railways also use carriage bogies with the bolster springs arranged longitudinally (fig. 3).

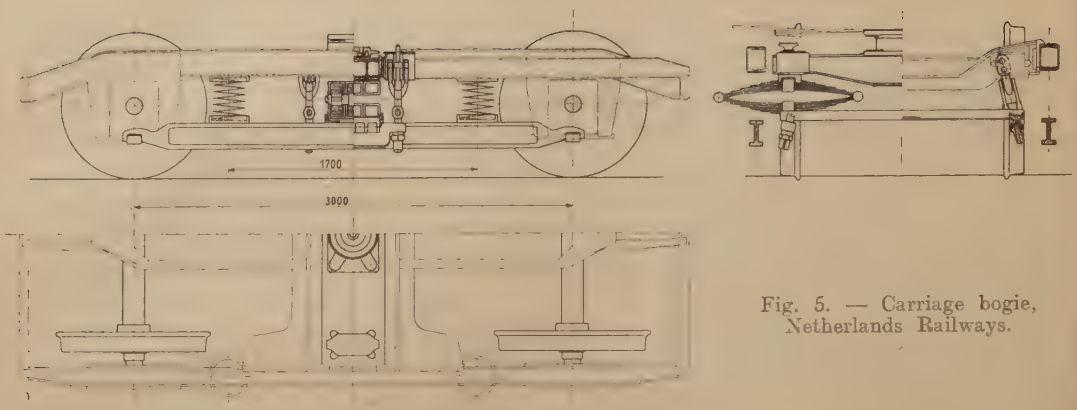


Fig. 5. — Carriage bogie, Netherlands Railways.

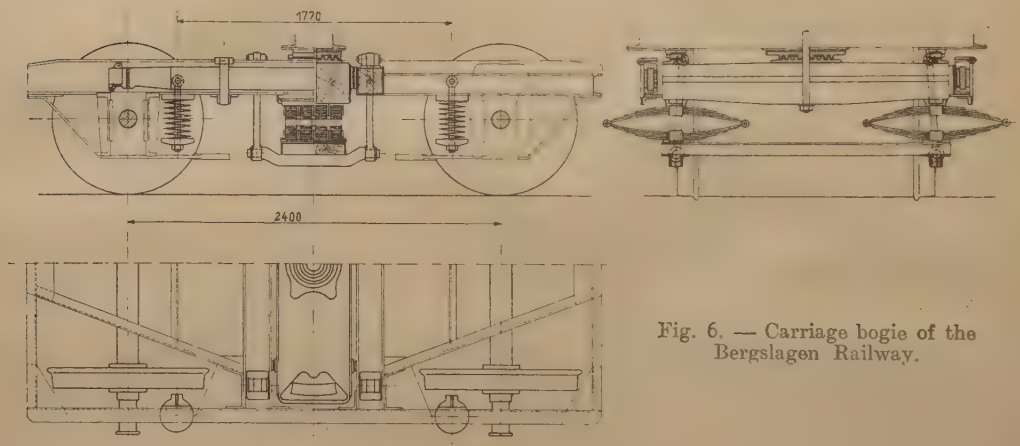


Fig. 6. — Carriage bogie of the Bergslagen Railway.

The narrow gauge [1.067 m. (3 ft. 6 in.)] is one of the reasons why the inclined swing links of the bolsters are placed outside the bogie frames. The two links on one side of the carriage each support a longitudinal beam outside the frame; the longitudinal beams of the two sides are connected together by round iron bars. They are placed under the longitudinal bolster spring on which the top bolster is carried. *Among all the bogies in question, the American type bogies and types derived therefrom are the only ones with triple suspension.* In this type of bogie the frame is carried by four helical springs bearing on two heavy equalising levers resting upon the two axle boxes. This method of support corresponds to the « Cs » type of support which we discuss in chapters II and III. The body of the carriage is carried on a transverse plane on the bogie. The equalisers ensure that the wheel loads on each side are equal. The ends of the equalisers are usually set upwards; in the most recent types these

bars are straight for their full length and are placed either above or below the axle boxes.

The advantage of the American bogies is that the vertical shocks can be damped out more completely thanks to the very flexible helical springs solely by internal friction. The equalising of the loads on the wheels on each side is another advantage of these bogies which is appreciated when running over slight differences in level of the track and defective surfaces (cf. Chapter III).

But as regards these advantages the construction generally adopted until recently had also some drawbacks. We mean the small wheel bases 2.15 m. (7 ft. 5/8 in.) to at the most 2.50 m. (8 ft. 2 7/16 in.), and the small bearing of the four helical springs, very close together longitudinally, for carrying the body and the bogie frame. This in fact is a consequence of the height of the equalisers, thanks to the turned up ends. Bogies with small wheel bases are necessarily less stable than those with long wheel bases. Furthermore the narrow wheel base facilitates end tilting of the frame, especially when starting and during heavy braking. Thus the ratio of the base of support to the wheel base for the old bogies of the German State Railways, is as 920 : 2150 = 1 : 2.34; for the cast steel bogie of the North of Spain Railways as 1250 : 2500 = 1 : 2; for the drop equaliser type of bogie of the Netherlands Railways, with the ends dropped particularly sharply as 1 600 : 2 500 = 1 : 1.56.

Among the 16 types of bogie with double suspension, 8 have pressed steel frames, and 6 frames of rolled sections; one administration — the North of Spain Railways — uses bogies of cast steel as shown in figure 4. The frame is a single casting in steel, with numerous lightening holes to reduce the weight. The longitudinal and tranverse girders are rectangular box girders, closed on all sides. The Madrid-Saragossa-Alicante Railways have also carried out tests with bogies of this kind.

The drawbacks of the small wheel base are avoided in the new bogies of the Netherlands Railways. The equalising lever is straight for its full length and attached under the axle boxes, so that it stresses them in tension. With these bogies it has been possible to get a wheel base of 3 m. (9 ft. 10 1/8 in.). The helical springs can under these conditions be brought further out. The base of support is 1.70 m. (5 ft. 7 in.) (1700 : 3000 = 1 : 1.76). For the rest, this bogie represents much progress well worthy of attention. The sole is formed of two U bars welded together at their open faces to form a closed box girder. Figure 5 shows this bogie.

In the same way in the case of the Bergslagen Railway Co's bogie shown in figure 6, the equaliser is straight, but in this case it is placed above the axle boxes. It is 145 mm. (5 45/64 in.) deep and can move in the U bars, 240 mm. (9 7/16 in.) high, of the longitudinal beam. This arrangement results in certain drawbacks such as the reduced play of the springs and the many openings required in the longitudinal girder. The base of support has become very large, with 1.77 m. (2 ft. 6 5/16 in.) for 2.40 m. (7 ft. 10 5/32 in.) wheel base. The ratio of the two dimensions which is 1 : 1.36 represents the best value of all the bogies with longitudinal equalisers with which we are dealing.

All double suspension bogies have bolsters with the springs arranged transversely. Besides inclined swing links, we also find on some of these bogies vertical links, for example on the drop equaliser bogies of the German State and the Danish State Railways.

Six-wheeled bogies are unusual. With the six-wheeled bogies of the German State Railways, fitted with equalisers, the body of the carriage is supported on a girder carried on the two bolsters. The links transmit the load to the frame

which passes it on to the axle boxes through the four helical springs and the drop equalisers.

The supports of the bolsters with transverse springs are generally of wood in the old carriages, to reduce noise; they are fitted with flat bars or rolled sections. In the most recent types of bogies the bolster supports are made of rolled sections (German State Railways), of pressed steel (Finland, Sweden), of cast steel (North of Spain Railways), or welded (Netherlands Railways). The play of the bolster in the longitudinal direction of the vehicle is limited within the necessary limits by rubbing plates and can readily be restored to the original amount after wear by renewing these plates.

On all triple suspension bogies, with the bolster springs arranged transversely, the *swing links* are inclined in varying degrees; thus for example, the links on the standard bogies of the old Prussian-Hessian State Railways with a length of 280 mm. (11 1/32 inches) have an inclination of 11° 30′; those of the alternating current rail motor coach of the German State Railways, only 170 mm. (6 11/16 inches) long, have an inclination of 10° 10′; and the links of the old bogies of the Swedish State Railways 665 mm. (2 ft. 2 3/16 in.) long, have only 2° 45′ inclination. The links are usually 250 to 350 mm. (9 27/32 to 13 3/4 inches) long. Unusually long links are in use on the carriages of the Northern States, Finnish State Railways 610 mm. (2 feet) Swedish State Railways 663 mm. (2 ft. 2 1/8 in.) and the Bergslagen Railways 606 mm. (1 ft. 11 55/64 in.).

On the bogies with double suspension the links are usually about 400 mm. (15 3/4 inches) long. The links of the new Dutch bogies are 425 mm. (16 3/4 inches) long and have an inclination of 11° 30′, and those of the Commonwealth bogies of the North of Spain Railways only 275 mm. (10 53/64 in.) in length, have an inclination of 12° 40′.

Vertical links sometimes known as parallel links (German State Railways, Danish State Railways) have been used on this type of bogie.

As will be shown in Chapter IV, greater centring forces can be obtained with inclined links than with parallel links of the same length.

The more the links are inclined and the shorter they are, the greater their centring power. In order to obtain with parallel links a sufficiently high centring force, their length must be very short [German State Railways 215 mm. (7 5/8 inches); Danish State Railways, 248 mm. (9 3/4 inches)]. The German State Railways report that the short parallel links have given better results on the American bogies than the inclined links.

The self-centring forces, from the information we have received, are generally small and vary between the limits of 300 to 1 000 kgr. (660 to 2 200 lb.); only in isolated cases does the final value increase to 2 300 kgr. (5 060 lb.) and this on bogies with bolsters having considerable transverse movement.

Nearly all the Administrations allow the bolsters to swing freely in the transverse direction, because experience acquired on their lines has shown that any restriction on the freedom of the body to oscillate, for example by fitting the bolsters with special buffers, detracts from the proper behaviour of the bogie and only results in hunting movements of the bogie being transmitted to the body. Only three Administrations (the former Bavarian and Saxon State Railways, and the Danish State Railways) use special bolster buffers with helical springs with some initial tension, which are fitted to both sides and act on the ends of the bolsters. The transverse movement of the bolsters is only 13 mm. (33/64 inch) on each side of the centre on the coaches of the former Bavarian State Railways; on most bogies it is

25 mm. (63/64 inch): German State Rys, Netherlands Rys, Dutch Indies State; it is as much as 30 mm. (1 3/16 inches) on the Finnish carriages and 40 to 50 mm. (1 37/64 to 1 31/32 inches) on the Spanish carriages. The solution of the question under consideration should obviously take into account the length restrictions. If the loading gauge is fouled by the side movement of the bolsters, this movement must be restricted.

The body of the vehicle is carried by the bogie on a *centre* bearing block (centre). These centres are of three different designs: the spherical bearing, the stepped bearing, and the flat disc bearing. From the purely static point of view (cf. Chap. III), single point support is the correct method as transverse inclinations of the body do not cause any alteration in the wheel loading. Another advantage of this arrangement is that it prevents skew at the bearing and consequently twisting of the body. The bearing on a spherical centre is equal to a single point support; but to facilitate lubrication and to get a lower surface pressure per square centimetre, it is generally made as a spherical annular support. These annular supports or bearings are found on 23 of the 45 types of bogies, namely on the carriages of the German State Railways, Finnish, Norwegian, Swedish and Dutch East Indies State Railways, as well as on the North of Spain and the Madrid-Saragossa-Alicante Railways. The radii of the spheres according to the information supplied or collected from other sources are 160 to 600 mm. (6 5/16 to 23 5/8 in.).

Except on some old types and rail motor coaches, the German State Railways no longer use the spherical bearing except in the case of coaches in the train ferry workings between Denmark and Sweden, a service in which changes of gradient on the bridges leading on to the ferry boats particulary stress the centre. All the other coaches of the German State Railways as well as of the the Netherlands Railways have stepped centres which give a more stable bearing and, thanks to the large surfaces in contact, damp out more quickly the hunting movements of the bogie.

The bogies of the Bergslagen Railway Co. (Sweden) are the only ones with single step discs of a special type. The load is carried on an annular surface. The same disc serves to centre the upper and lower bearings. The concentric flanges of the disc on the two sides only appear to act as reservoirs for holding the grease.

As far as the information supplied makes it possible to judge, the body is solely carried on the centres on nearly all carriages.

Side friction blocks or auxiliary bearings are in use on all carriages, but these fittings only act as bearings in the event of lateral oscillations of the body or transverse inclinations of the bogie in the direction of the side bearings equalling the play of 0.5 mm. to 1.5 mm. (5/256 to 1/16 inch.), which exists in the central position. If the play were smaller or even nil, hunting movements would be very effectively wiped out, but the ability of the bogie to run over the track with the minimum of guiding action would be lessened. The inevitable consequence would be considerable wear of the sides of the rails and the safety of running would suffer. In on the other hand the play is too great, the body causes unpleasant blows on the bogies. Endeavours are frequently made to cure this defect which is always caused by too little or too great play at the side friction blocks. With this object the coaches of the Danish State Railways are fitted, so far as can be seen from the design, with roller side bearings, and those of the Bergslagen Railway with conical roller bearings, i. e. with bearings in which the frictional resistance

is notably less than that of the plain bearings. These bearings have no vertical play, they are taken as constantly carrying the load in common with the centre. Now the distribution of the weight on three non-elastic supports is known to be statically indeterminate. It may happen that the load is distributed unilaterally (on defective sections of the track and on transitions; cf. Chap. III). The same thing naturally can also occur with slotted bearings.

In accordance with § 86 of the German Railway Association's recommended practice (1), the bogies must be able to turn sufficiently for the bogie vehicles to run without difficulty through curves of 150 m. (7 1/2 chains) radius. It is recommended that the bogies on passenger bogie carriages should be so arranged that the vehicle can run through curves of 100 m. (5 chains) radius.

The angle a bogie describes when running through a given curve depends upon the radius of the curve, upon the distance between the bogie centres, upon the bogie wheel base and upon the play between the wheels and the track. The angles of rotation of the bogies as a rule are only limited by the safety chains. Several Administrations such as for example the Dutch East Indies State Railways and the Bergslagen Railway do not limit the angle of rotation in any way. Some Administrations have not given any information upon this subject.

The wheel base of double suspension drop equaliser bar type bogies, of the pattern in general use up at present, is 2.15 m. (7 ft. 5/8 in.) (German State Railways) to 2.50 m. (8 ft. 2 7/16 in.) (Netherlands, Madrid-Saragossa-Alicante, North of Spain), while that of the new bogies of the Netherlands Railways is 3 m. (9 ft. 10 1/8 in.). The normal type of bogies with triple suspension have a wheel base of 2.20 m. (7 ft. 2 5/8 in.) (Finland, Dutch East Indies) to 3.50 m. (11 ft. 5 25/32 in.) (former Bavarian State Railways), but the most usual value is ∓ 2.50 m. (8 ft. 2 7/16 in.).

The six-wheeled bogies of the German State Railways have a wheel base of 3.60 m. (11 ft. 9 3/4 in.); the four-wheeled heavy type II Görlitz bogies have also a 3.60-m. wheel base, and consequently the longest wheel base of all four-wheeled bogies mentioned in the replies.

The distance between bogie centres varies between the limits of 9.20 m. (30 ft. 2 7/32 in.) [Dutch East Indies, 1.067 m. (3 ft. 6 in.) gauge and 16.18 m. (53 ft. 1 in.) (the coaches of the Rheingold express of the German State Railways)]. The most usual dimension are 12 to 15 m. (39 ft. 4 1/2 in. to 49 ft. 2 1/2 in.).

The overhang and the width of the coaches have values which depend upon the width allowed by the loading gauge and the distance between centre pins. All carriage builders know how these dimensions are arrived at. An excessive overhang adversely affects the stability of all coaches, especially at high speeds. With few exceptions (for example the former Bavarian State Railways) the relatively small wheel bases of 2.15 m. to 2.50 m. (7 ft. 5/8 in. to 8 ft. 2 7/16 in.) have therefore been selected for the bogies of passenger carriages because in this way the overhang can be kept small.

The information we have obtained shows, however, that considerable importance is now being attached to the wheel base, not to the overhang, without however underestimating the effect of excessive overhang.

In the remarks with which we began this paragraph No. 3 we pointed out that bogie vehicles are considered as the most suitable type for high speeds. We thought that it was necessary first of all to give a description with our comments, limiting ourselves in the spirit of the task we have undertaken, to tech-

(1) Technische Vereinbarungen.

nical questions relating to the running. Out of the 10 Administrations who replied to the questionnaire, 6 declared that they use bogie carriages exclusively for trains the speed of which reaches or exceeds 80 km. (50 miles) per hour. These railways are : The German State Railways, Finnish State, Norvegian State, Danish State, Dutch East Indies, and Madrid-Saragossa-Alicante. The Netherlands Railways use bogie stock exclusively for trains running at a speed of 90 km. (56 miles) an hour and over.

The North of Spain Railways permit as a special measure four-wheeled vehicles to be added as strengthening vehicles to bogie carriages in express trains. *The Swedish State Railways and the Swedish Bergslagen Railway alone use four-wheeled vehicles with radial axles on express trains regularly. The Swiss Federal Railways advised us* subsequently to the tables being compiled, *that they also allow four-and six-wheeled vehicles behind all express locomotives,* provided such vehicles have a wheel base of at least 60 % of the length of the frame. In addition the wheel base of goods wagons must be at least 5 m. (16 ft. 5 in.), and that of passenger coaches, luggage vans and post office vans of at least 6 m. (19 ft. 8 1/4 in.). In addition the tare of the passenger coaches must be at least 14 tons, and that of the post office vans and parcels vans, 16 tons.

The four-wheeled vehicles of the two Swedish Administrations are vehicles with radial axles with 7.5, 8 and 9 m. (24 ft. 7 9/32 in., 26 ft. 3 in. and 29 ft. 6 3/8 in.) wheel base, the play of the radial axles in the longitudinal line of the vehicle is 2.5 $\times$ *a* millimetres, *a* being the wheel base. The rigid wheel base of four-wheeled vehicles of the North of Spain is 7 m. (22 ft. 11 5/8 in.). The Swiss Federal Railways use radial four and six-wheeled vehicles with varying wheel bases up to 9.20 m. (30 ft. 9 3/4 in.); the radial axles have a side play of 5 mm. (13/64 inch) and a longitudinal play of 25 mm. (63/64 inch) about the centre. The owing Administration reports that these vehicles can run through curves of 90 m. (4 1/2 chains) radius. The middle pair of wheels of the six-wheeled vehicles is carried in a frame which is free to move and allows 70 mm. (2 3/4 inches) side play on each side of the centre.

The train unit of the Swedish State Railways consisting of 4 bodies carried on *3 Jakobs bogies of the Görlitz type* and on *2 radial axles* at the two ends is an up-to-date solution of the problem of reducing the tare weight per seat to a minimum. The ends of the two adjacent bodies are carried on hemispherical couplings on the same bogie.

The guiding pairs of wheels, i. e. the pairs of wheels which decide the position of the coach or of the bogies and which guide the vehicle on the straight and on curves are the extreme outer pairs of wheels on all the bogies and on all the rigid wheel vehicles. With the exception of the few carriages with radial wheels and axles they are always mounted rigidly. We have already said everything necessary about the distance of the guiding pairs of wheels when speaking earlier about bogies and rigid wheel base vehicles. It must be mentioned here that the side play of the rigid wheels is generally less than 5 mm. (13/64 inch), according to the information supplied. The German State Railways, for example, prescribe a side play of $\pm$ 0.5 mm. (5/256 in.) on the axle box brass and $\pm$ 1 mm. (5/128 inch) in the axle box guide. The Finnish State Railways specify a side play on both sides of the centre line of 2.5 mm. (13/128 inch) on the brass and 1 mm. (5/128 inch) in the axle box guide.

The *axle load* is 6.75 t. (6.64 Engl. tons) in the case of the Dutch East Indies State Railways [1.067 m. (3 ft. 6 in.) gauge]; it varies from 9 to 12 t. (8.85 to 11.8 Engl. tons) for most of the standard gauge bogie carriages and rigid-wheeled

vehicles; the most heavily loaded pairs of wheels are those of the German « Rheingold » coaches with 13.97 t. (13.75 Engl. tons). In the case of rail motor cars axle loads are found of 12 t. (11.8 Engl. tons) (Dutch East Indies), to 18.4 t. (18.1 Engl. tons) (the alternating current rail motor coach of the German State Railways).

The smallest wheel diameter on high-speed vehicles is 764 mm. (2 ft. 6 5/64 in.) on the Dutch East Indies Railways [1.067 m. (3 ft. 6 in.) gauge]. Then comes, for standard gauge, Sweden with 800 mm. (2 ft. 7 1/2 in.) in the case of vehicles with radial axles, Norway with 910 mm. (2 ft. 11 7/8 in.), Denmark and the Bergslagen Ry. (Sweden), with 966 mm. (3 ft. 2 in.), and in the case of the broad gauge, Finland with 960 mm. (3 ft. 1 25/32 in.). The other Administrations, including the Spanish Railways, use wheels 1 m. (3 ft. 3 3/8 in.) in diameter on the tread.

An important means of damping out excessive oscillations of the coaches, and consequently of improving the safety of running, is the *close coupling* of the vehicles. All the Administrations, with the exception of the Dutch East Indies State Railways which have automatic couplings use screw couplings, with right and left hand threads for coupling the vehicles together tightly. Special devices designed to tighten the coaches together more closely after the couplings have been screwed by hand are not used by any Administration. When tightening up the coupling, a lever is used, the length of which from the axis of the screw to the extreme end is, for example, 230 mm. (9 1/16 inches) on the Norwegian State Railways, 275 mm. (10 53/64 inches) on the German State Railways, 470 mm. (18 1/2 inches) on the North of Spain Railways; on some railways also this lever has a heavy ball weight at its end (Madrid-Saragossa-Alicante Railways and North of Spain Railways). From the information supplied the buffers are generally fitted with compensating gear so that the pressure on both the buffers is equal even on curves, and additional guiding pressure due to the pressure of the buffers is avoided. The buffer pressure experienced in normal working from the information given in the replies amounts to between 300 and 2 700 kgr. (660 and 5 950 lb.); it depends upon the length of the levers of the screw couplings, the pitch of the thread, and the effort exerted by the shunter when coupling up.

In order to damp out vibrations and lessen noise, some Administrations use rubber pads at various points. For example, on the German bogie coaches fitted with Görlitz type bogies *rubber has been used successfully between the top and bottom centre castings,* as well as between the bolster and the side friction blocks. As long ago as 1906, the Java bogie carriages were fitted with two rubber pads between the eight leaves of a spring; this test was not satisfactory and was soon abandoned. On the other hand, at the present time fresh tests are being made with a rubber pad known as « Resonite » under the spring buckles. The Netherlands Railways fit on their electric stock rubber pads at both ends of the helical springs. The rail motor coaches were fitted with rubber pads at the points of support of the laminated springs between the main frame and the top centre casting, and between the main frame and the top side friction blocks, but these have not always been particularly satisfactory, and have been replaced by felt pads. The Norwegian State, the Bergslagen Railway and the Danish State Railways use felt pads between the body and the frame while the North of Spain uses rubber body blocks.

In the case of express train coaches, the different Administrations do not usually lay down any maximum speed. The carriages considered in this report can run at any speed allowed over the different lines. The maximum speed is

only limited in the case of the rail motor coaches, for reasons relating to starting.

The whole of the *details of the spring gear* are of particular importance as regards good running and immunity from derailment. It may be remembered that all the bogies with triple suspension gear are not fitted with equalising gear whereas on the other hand all the bogies with double suspension gear are fitted with longitudinal compensating gear by means of equalising levers of either the drop or straight pattern. The four-wheeled rigid wheel base vehicles have generally single springing but the radial axled vehicles of the Bergslagen Railway have double suspension. The following details of the spring gear may be mentioned :

The bogie bearing springs properly speaking are 900 mm. to 1.25 m. (2 ft. 11 1/2 in. to 4 ft. 1 13/64 in.) long, 65 mm. to 120 mm. (2 9/16 in. to 4 23/32 in.) wide and consist of 5 to 12 plates 10 to 16 mm. (25/64 in. to 5/8 in.) thick. Consequently the strength of the springs also varies between wide limits : there are stiff or hard springs with a strength of 111 kgr. per mm. (5 656 lb. per inch), and soft springs with only 30.3 kgr. per mm. (1 696 lb. per inch). The deflection of these springs with the vehicles empty is 46 to 169 mm. (1 13/16 in. to 6 43/64 in.). The stress on these springs is nearly always very high; for example it is 101 kgr. per mm² (64.1 Engl. tons per sq. inch) on the Görlitz bogies.

The bearing springs on carriages carried on radial axles are particularly long [2.08 m. to 2.13 m. (6 ft. 9 29/32 in. to 6 fr. 11 7/8 in.)] and as soft as possible, through the use of a large number of thin plates. The plates are 90 to 100 mm. (3 17/32 in. to 3 15/10 in. wide, 12 to 14 mm. (15/32 in. to 35/64 in.) thick, and 11 to 14 in number. From the information supplied, the strength of these springs is 21.3 kgr. to 21.65 kgr. per mm. (1 193 to 1 212 lb. per inch). which makes it possible to get a deflection of 169 mm. (6 43/64 inches) on the Bergslagen Railway carriages when empty, for example.

The secondary suspension of carriages with triple suspension is arranged to come between the laminated bearing springs and the bogie sole bars and consists of either volute springs (on the former Bavarian and Baden State Railways) or coiled springs, with elliptical, circular, rectangular, square or special section coils. The strength of these springs is 53.4 to 133 kgr. per mm. (2 990 to 7 448 lb. per inch) the deflection is 18 to 98.1 mm. (3 45/64 to 3 7/8 inches) (this latter figure only applies to volute springs) and the stress is as high as 96 kgr. per mm² (60.9 Engl. tons per sq. inch.).

The springs between the drop type equalisers and the bogie sole bars in bogies with double suspension are all coiled springs made of circular or square section bars; generally three but sometimes two springs of different diameter and strength are inserted one inside the other. These springs have a strength of 36.4 to 83.4 kgr. per mm. (2 038 to 4 670 lb. per inch), deflect up to 126 mm. (5 inches) under loaded vehicles, and are stressed to, for example, 65 kgr. per mm² (41.2 Engl. tons per sq. inch) on the German bogies with drop equalisers.

The *transverse bolster springs* consist of three or four groups of full elliptical springs; the bolster springs of the six-wheeled bogies alone are arranged in two groups. These springs fully occupy the space available; they are 762 mm. to 1.02 m. (2 ft. 6 in. to 3 ft. 4 5/32 in.) long, 70 to 120 mm. (2 3/4 in. to 4 23/32 in.), usually 90 mm. (3 17/32 in.) wide, and are built up of 2 × 4 to 2 × 8 plates, 7 to 16 mm. (9/32 in. to 5/8 in.) thick. With a strength of 37.8 to 78.2 kgr. per mm. (2 116 to 4 379 lb.), the deflection is 47 to 140 mm. (1 27/32 to 5 1/2 in.), the stress being as high as 119 kgr. per mm² (75.5 Engl. tons per sq. inch).

The longitudinal bolster springs on the Görlitz bogies are 2.05 m. (6 ft. 8 45/64 in.) long on the heavy type II, 1.80 m.

(5 ft. 11 in. long on the light type III, and are made of 120 × 16 mm. (4 23/32 × 5/8 in.) spring steel, and have 7 to 9 plates. With a strength of 54 to 62.5 kgr. per mm. (3 024 to 3 500 lb. per inch), the deflection is 131.5 mm. to 152 mm. (5 5/32 in. to 6 in.); the stress is only 66 to 73 kgr. per mm² (41.9 to 46.3 Engl. tons per sq. inch) and is therefore not particularly high.

With the double or triple suspension, the total deflection of the springs on the bogie carriages when empty is 85 to 264.5 mm. (3 11/32 in. to 10 13/32 in.), and when the vehicle is fully loaded 110 to 310 mm. (4 11/32 in. to 12 3/16 in.). The German vehicles are particularly well sprung, while the Finnish vehicles have a particularly hard suspension gear.

We investigate the properties of laminated and coiled springs in Chapter III, when dealing with the springing of locomotives. Investigations and measurements relating to the question as to how the springs behave under load whilst running and what are the oscillations of the bodies relatively to the bogie sole bars, and to the axles, are actually being carried out in the mechanical engineering headquarters office of the German State Railways; they have not yet been completed. The Dutch East Indies State and the Netherlands Railways are also carrying out similar tests. As to the results, it is only possible to report the two following facts at the moment : laminated springs have only given when running, the deflections due to changes in the static load; secondly, practically no vertical sinusoidal vibrations of the springs have been observed. It is expected that similar results will be obtained from the oscillograms being taken at the Higher Technical School of Berlin by the German State Railways with the assistance of the Heinrich Hertz Institute for the study of vibrations, to determine the changes in the pressures of the guiding wheels of the vehicles while running over irregular surfaces of the track. On the curves, the action of the centrifugal force shows itself as an additional static load.

Even the best design is no guarantee of the stability of the vehicle at the long run; constant care must be given in service to keep in good order and without excessive wear the details upon which the safety of running depends. For this reason, on the German State Railways, bogie carriages running in express trains are examined very carefully every 2 to 6 weeks at their home station and certain parts checked to see that the dimensions specially marked down on the diagram sheets are not exceeded, for reasons of safety, either above or below the given limits; when doing this work no part of the vehicle is taken down nor is the vehicle lifted. These checks make it possible to ascertain if a spring has lost its elasticity or is broken and make it certain that the body is well supported, that the bogies can run through curves without difficulty, and that the tyres are in proper working condition.

On the German State Railways, the bogie vehicles used on express trains are sent to the shops after 100 000 km. (62 000 miles) or every six months at the latest. All parts are examined, especially those upon which safety of running depends and checked to see that the wear has not exceeded certain specified limits. Parts which are much worn and damaged, are either made good or replaced. When repairs have been completed, the limiting dimensions must not be exceeded anywhere.

At each examination the body is tried over to make sure that it can not be twisted by the forces acting on the buffers and centres, that the bogies have no tendency to strike the rails on a particular side and that the body supports are well centred on the bogies and consequently that the bogie springs are uniformly loaded.

The other Administrations, excepting the Swiss Federal Railways and the Finnish State Railways usually attach less

importance to accurate checking of the dimensions, to judge by the little information we have received on the subject. All that is done is to inspect the parts affecting the safety of running and to take them down if need be, to repair or renew them.

The examination takes place after a given distance has been run or after a given period of service. Until recently on the German State Railways this examination was made after a mileage of 75 000 km. (46 600 miles) and at the latest at the end of six months. By taking care that the repair work was done carefully in the shops and that running repairs were carried out properly, it has recently been found possible to increase the distance to 100 000 km. (62 000 miles); but the period in service is still limited to six months. Table 2 indicates the time in service and the mileage between repairs fixed by the other administrations.

TABLE 2.

ADMINISTRATION.	Distance run between two examinations of bogies.	Period in service between two examinations of bogies.
German State Railways	100 000 km. (62 000 miles).	1 year at most.
Finnish State Railways	...	1 year.
Norwegian State Railways	40 000 km. (25 000 miles).	6 months as a rule.
Swedish State Railways	...	4, 6 or 12 months, depending on fatigue of carriage.
Bergslagen Railways	...	4 months.
Dutch Indies State Railways . . .	80 000 km. (50 000 miles).	So long as the coach has not run 25 000 miles, the period may reach 3 years.
North of Spain Railways	...	2 years (after 1 year intermediate examination; every 6 months the body is lifted.
Swiss Federal Railways	...	6 months for carriages in international service, 2 years for four, six and eight-wheeled vehicles in home service.

It would not be correct to conclude that those Administrations who prescribe a short period of service or a small mileage between two repairs maintain their vehicles in better condition. The truth is that when vehicles rarely come into the shops more careful supervision and better maintenance in service is required.

Summing up the different details relating to high speed coaches, details which we have had to give for completeness, we find the following as regards question III :

The great majority of vehicles for high speeds are bogie vehicles, and in these the types of bogie with triple suspension

and body bolster springs arranged transversely are at the present time the most widely used, the type with longitudinal springs being still a new one. The American bogie with double suspension continues to be used, but at the same time an endeavour is being made to increase the small wheel base which has had to be used up to the present by a rational arrangement of the equalisers. The users of one or the other type agree that to improve the running, wheel bases of at least 3 m. (9 ft. 10 1/8 in.) and soft springs must be used.

Four and six-wheeled vehicles can only be used without hesitation for high speed, when they have a long wheel base and the wheels carry sufficiently high loads. From the point of view of stability when running, these vehicles are not as good as bogie coaches, and for this reason alone, most of the Administrations prefer not to use them on express trains.

CHAPTER II.

Method of support of locomotives and tenders and position of their centre of gravity.

The way the bogie springs are carried on the bogie and their relative strength as compared with the other engine springs, as well as the manner in which the main frame is supported on the bogie and in which it is carried itself, exercice during the running of bogie locomotives over irregular track a great influence on the proportional modifications of the pressures of the guiding wheels of the bogie. For this reason it is necessary to consider very closely the design of the bogies and to bring out the distinctive features, i. e. those which, according to the design, affect the variations in the pressure of the directing wheels. We use with this object the « type of support characteristic » C which we will first of all explain.

If the main frame is carried on the bogie on two side bearings, we will use C as an abbreviation to denote this method; if there is a central spherical centre it will be denoted by C*m*; if the method of supporting the main frame is approximately like that of C*m* the abbreviation C*m'* will be used. The letter *e* will be added if the bogie is carried by four individual springs; *l* if the four springs are articulated in pairs by the longitudinal equalisers; *s* if the load is transmitted by two longitudinal equalising springs to the two pairs of wheels of the bogie; *v* if there is an equaliser at the leading end; *h* if the compensating equaliser is placed at the trailing end. The characteristics C*e*, C*l*, C*s*, C*v*, C*h*—C*me*, C*ml*, C*ms*, C*mv*, C*mh*—C*m'e*, C*m'l*, C*m's*, C*m'v* and C*m'h* will also be used.

First of all a description will be given of the methods of support used on steam locomotives, showing in each individual case the essential points of the design by means of the « support characteristic » and *in italics* at the end a summary of the methods of support selected and the frequency with which they are found.

The same procedure will then be followed for electric locomotives and tenders.

Some general preliminary remarks should be made.

One alone out of the 30 locomotives in question is fitted with a transverse equaliser and that is the 4-8-2 of the North of Spain Railways; the odd number of bearing points (3 and 5) is obtained on all the other locomotives by the main frame being supported at the middle of the bogie. It should be especially noted that in no case is a bogie not fitted with a transverse equaliser.

If the main frame is carried on a flat bearing at the middle of the bogie, the main frame transfers its transverse inclination to the carrying surface of the

TABLE 3.

Method of supporting the 30 steam locomotives mentioned in chapter I.

	1	2	3	4	5	6	7
	Units.	Wheel arrangement.	Carried				
			in two planes.		in three planes.		
			on				
			3	4	4	5	6
			points.				
a)	9	4-6-0	1 DR 2 Fin 1 NS 1 BJG	1 N	...	...	2 DR 1 SpN
b)	11	4-6-2	1 S	4 DR	...	1 DR	2 DR 2 NI 1 SpN
c)	2	4-6-4	...	...	1 NS	...	1 NI
d)	2	4-8-0	...	...	...	1 SpN	1 SpN
e)	2	4-8-2	1 SpN 1 MZA	...	...	...	...
f)	1	4-8-4	...	...	...	...	1 SpN
g)	1	2 8-0	...	...	...	...	1 SpN
h)	2	2-8-2	1 SpN	...	...	...	1 DR
Total	30	units	9 (30 %)	5 (16.7 %)	1 (3.3 %)	2 (6.7 %)	13 units (43.3 %)

Note : For explanation of abbreviations, see table 1, p. 2262.

bogie and consequently to the frame thereof. This method of support is therefore equivalent to the main frame being carried on two bearings placed to the left and to the right of the longitudinal axis of the vehicle, i. e. to a « bilateral » support of the main frame. As regards variation in pressure of the wheels on irregular surfaces of the track, in this case it is unimportant whether the side bearings rest on flat or hemispherical surfaces. The only difference

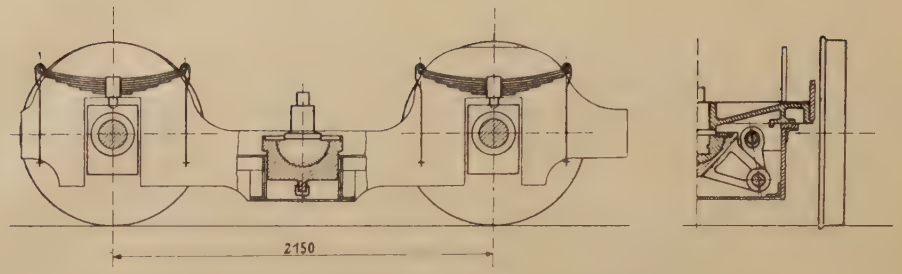


Fig. 7. — Bogie of series 17[7] locomotive, German State Railways (former Saxon State Railways).

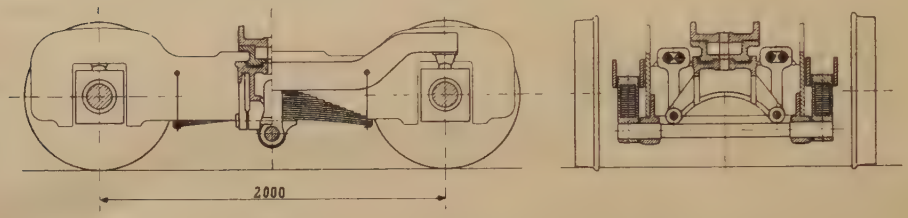


Fig. 8. — Bogie of series H[9] locomotive, Finnish State Railways.

is that in the case of hemispherical surfaces, the bearing takes place upon what is called a « transverse fulcrum », i. e. when seen in elevation, rotary movements of the main frame and of the bogie about a transverse knife edge are possible.

If, on the other hand, the main frame is carried by the bogie through a spherical centre, the changes of the guiding wheel pressures through the irregular surfaces of the track are not so unfavourable. This question is dealt with in a more detailed manner in Chapter III.

The design of the parts of the running gear having to guide the leading end of the locomotive is an element of great importance in the question we are dealing with. By making use of table 3 we can consider these details in the order of succession of colums 3 to 7.

Column 3, line a). — Among the 5 locomotives with the (4-6-0) wheel arrangement, only one has a central spherical bearing : this is an old Saxon locomotive (type XII HV). The centre is carried on a bolster with inclined links (for the centring force of this bolster see Chapter IV), the frame of the bogie is carried on 4 independent laminated springs (see fig. 7). This method of support has been given the characteristic C*me*. The four other locomotives are carried on a centre plate bearing; they are carried therefore, to be exact, on two flat bearings very close together, so that in reality these locomotives should be entered in column 4; we have mentioned them here because these two imaginary supports are situated at a short transverse distance and because the other conditions of the method of supporting the bogie, the inevitable drawbacks of the flat bearing, are not important in this case as regards the classification of the method of support from the point of view of the statical value. Actually among these four loco-

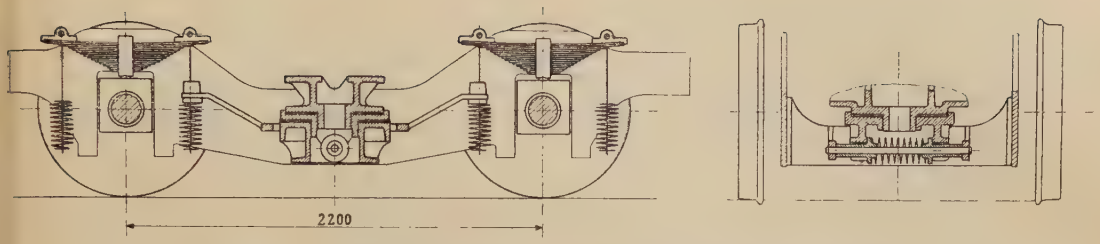


Fig. 9. — Bogie of series 3900 locomotive, Netherlands Railways.

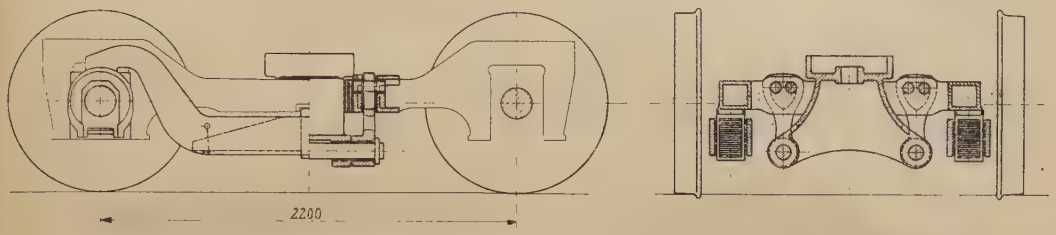


Fig. 10. — Bogie of serie 1700 locomotive, Madrid-Saragossa-Alicante Railways.

motives, two (Finland) are fitted with bolsters with parallel links and with two longitudinal springs for the longitudinal drop type equaliser (see fig. 8, characteristics *Cm's*); one locomotive (Netherlands) has helical coiled springs with a longitudinal equaliser between the four laminated bearing springs of the bogie (fig. 9, characteristic *Cm'l*), and one other locomotive (Bergslagen Railway) has a bolster with parallel swing links, and so far as can be seen from the designs sent in, a longitudinal equaliser between the laminated bearing springs of the bogie, characteristic *Cm'l*.

Column 3, line b). — Among the 11 locomotives of the (4-6-2) type one only (Swedish State) is carried on a centre flat bearing, on a bolster with parallel links fitted with roller bearings. The four laminated bearing springs are connected by longitudinal equalisers, characteristic *Cm'l*.

Column 3, line e). — On the 4-6-2 locomotives of the North of Spain Railways, the main frame is carried on a spherical bearing on the bogie, which has a bolster with inclined links and four-point bearings. The four laminated bearing springs each with two auxiliary coiled springs are loaded indirectly; characteristic *Cme*.

The 4-8-2 locomotives of the Madrid-Saragossa-Alicante Railways has its main frame supported on a plain centre bearing which rests on a bolster fitted with parallel links with roller bearings and transmits the load to two longitudinal equalising springs with drop equalisers, figure 10, characteristic *Cm's*.

Column 3, line h). — The 2-8-2 locomotive of the North of Spain Railways is supported by the three leading axles at a single point, through a transverse equaliser in front of the leading pair of coupled wheels, and at two points on the

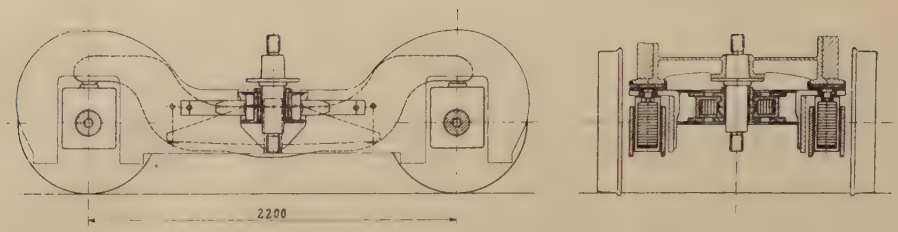


Fig. 11. — Bogie of standard locomotive, German State Railways.

three trailing pairs of wheels. The details are very like those of bogies of American design. The load on the leading pony truck is carried on a bolster with parallel roller bearing links. Short cylindrical helical springs transmit the pressure at the bolster to the pony truck bearings.

Column 4, line a). — No details of the bogie of the 4-6-0 Norwegian locomotive were received.

Column 4, line b). — Locomotives of the 4-6-2 type with four-point suspension have not been mentioned except by the German State Railways. The four-wheeled bogies of the three standard locomotives included therein are derived from the old standard Prusso-Hessian pattern which, according to the German State Railways, gave good results. These bogies have two longitudinal compensating springs on drop equalisers upon which the main frame rests through flat slides on two supports 1.05 m. (3 ft. 5 23/64 in.) centre to centre apart, figure 11, characteristic C*s*. The bogie centre is cylindrical. It can slide freely up and down, without taking up any of the load, in the bearings, and the cast steel centre, with bronze linings arranged transversely to the longitudinal axis, causes it to take part in absorbing the running resistance of the bogie and the guiding effort of the centre. The guiding box is cylindrical at the middle of the centre for a length of 60 mm. (2 23/64 inches) and is tapered 1 in 15 at both ends, i. e. at the top and at the bottom. In this way the bogie can take up the longitudinal and transverse inclination relatively to the main frame without constraint and without the cylindrical centre being subjected to bending.

The fourth of these locomotives is of the old Bavarian type. Its bogie is carried on four points or « transverse knife edge support [two centres 960 mm. (3 ft. 1 1/4 in.) apart laterally], with hemispheric bearing surfaces at the centres) and the bogie is guided in a sperical bearing, characteristic C*e*. This locomotive and the 4-6-2, series 01, of the German State Railways, are used in Chapter III as examples for the calculations.

Column 5, line c). — The 4-6-4, series 6100 tank engine of the Netherland Railways, is supported, as all the following locomotives, in three transverse planes. The two bogies carry the main frame on centre bearings; the frame is therefore supported on four points (lozenge). No information has been given on the subject of the spring gear and the equalisers.

Column 6, line b). — The bogie of this old Saxon 4-6-2 locomotive is the same type as that of the old Saxon 4-6-0 type already described, column 3, line *a*), characteristic C*me*.

Column 6, line d). — The method of construction of the bogie of the 4-6-0, series 4300, locomotive of the North of Spain Railways corresponds to the characteristic C*m's*. The central flat bear-

ing has the exceptionally large diameter of 620 mm. (2 ft. 3/8 in.).

Column 7, line a). — The two 4-6-0 German locomotives are of the former Prusso-Hessian type; their bogies correspond to the type described in column 4 *b*). Characteristic C*s*.

The 4-6-0 locomotive of the North of Spain Railways has a bogie of the characteristic C*e*.

Column 7, line b). — The two 4-6-2 locomotives are locomotives of the old Baden and Wurtemberg types; their design corresponds to the characteristic C*e*.

The two 4-6-2 locomotives of the Dutch East Indies State Railways [1.067 m. (3 ft. 6 in.) gauge] have bogies with the characteristic C*e*.

In the same way the bogie of the 4-6-2 locomotive of the North of Spain Railways has the characteristic C*e*.

Column 7, line c). — The 4-6-4 locomotives of the Dutch East Indies State Railways (1.067 m. gauge) have bogies with the characteristic C*s*, i. e. with the main frame supported bilaterally on transverse knife edges and with longitudinal drop type equalisers.

Column 7, lines d) *and* f). — The bogies of each of the 4-8-0 and 4-8-4 locomotives of the North of Spain Railways support the main frame on a cross knife edge, the slightly spherical centre of the bogie is guided in a cylindrical cast steel guide, and has four-point suspension, corresponding therefore to the characteristic C*e*.

Column 7, line g). — The main frame of this 2-8-0 locomotive of the North of Spain Railways is carried on the two sides of the pony truck. Each support is fitted with two rollers which run up inclined planes when passing through turn outs. The case of the load supported on a single pair of wheels does not come within the scope of these comparisons. On all the other steam locomotives mentioned, the loads on the first and second pairs of wheels are mutually interdependent.

Column 7, line h). — This 2-8-2 locomotive is of the Saxon type with an old type Krauss-Helmholtz bogie (fig. 12) The method of support corresponds to the characteristic C*e*.

The above means that *of 30 steam locomotives mentioned in the replies, 9 (30 %) are suspended on three points, 6 (20 %) on four points, 2 (6.7 %) on five points, and 13 (43.3%) on six points.*

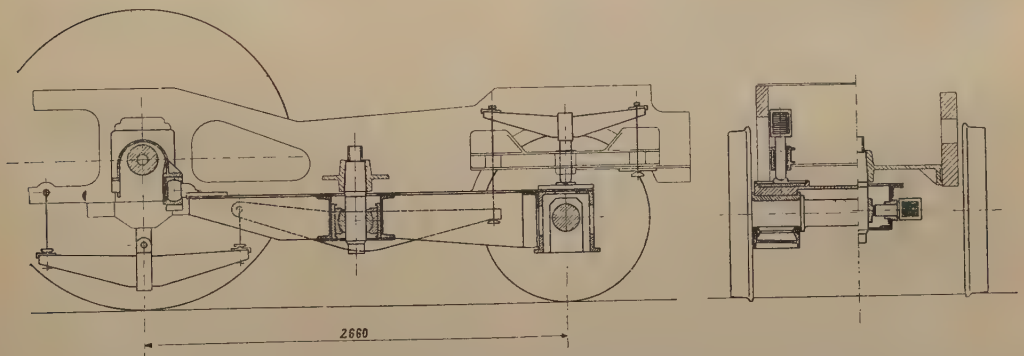


Fig. 12. — Krauss-Helmholtz bogie of series 19^{0} locomotive, German State Railways (former Saxon State Railways).

14 locomotives are supported in two planes, and 16 locomotives in three planes.

Among the 30 locomotives, 27 (90 %) have four-wheeled carrying bogies, one locomotive is fitted at the front end with the Krauss-Helmholtz bogie, and 2 locomotives have a guiding truck.

On the bogies of 28 locomotives, 10 (35.7 %) correspond to the characteristic Ce, *three (10.7 %) to the characteristic* Cme, *six (21.4 %) to the characteristic* Cs, *four (14.3 %), to the characteristic* Cm's *and three (10.7 %) to the characteristic* Cm'l. *In the case of two locomotives the support characteristics are not known.*

None of the bogies have transverse equalisers at the leading end, so that the characteristic Cv *is not found on any steam locomotive.*

Method of support of electric locomotives.

Among the 14 electric locomotives for express trains (see table 4 below), there are only 9 with guiding bogies; these have 2 pairs of carrying wheels on 6 locomotives, and one pair of carrying wheels and one pair of driving wheels on three locomotives. Among the steam locomotives mentioned in the replies, 90 % had guiding bogies with two pairs of carrying wheels; here only 64 % of the locomotives are fitted with bogies, and 43 % of the locomotives mentioned have bogies with two pairs of carrying wheels. This is due to the fact that the electric locomotive must be designed so as to be able to run equally well in both directions, whereas the steam tender locomotive is generally only required to comply with the condition that the running gear shall properly guide the engine for one direction of running.

As regards the method of supporting the electric locomotives, transverse equalisers placed at the leading end are a very usual form of construction to ensure that the static load shall be the same on both the left and right wheels of the leading directing pair of wheels. Bogies with carrying pairs of wheels of the 2 C 2, 2 BB 2 and 2 Do 1 locomotives (series E 21 and E 21[5]) of the German State Railways are fitted with transverse equalisers on their leading carrying wheels; in the same way the trucks of the 2 Do 1 locomotives (series E 21 and E 21[5]) and 2 D 1 (series E 50) of the German State Railways have transverse equalisers. Consequently on the five types of locomotives of the German State Railways which have one or two four wheeled bogies, four types have in both directions of running transverse equalisers above the end pairs of wheels.

In the case of 5 locomotives the main frame is supported on their carrying bogies on a transverse knife edge (two lateral supports on hemispherical bearings); in the case of the 2 Co-Co 2, series 7200, locomotives of the North of Spain Railways alone the frame is supported on the carrying bogies through a centre pin, i. e. on the centre of the transverse axis.

First of all we will call attention to the noteworthy features of the method of support of these locomotives with the aid of table 4 below, following this time the order of the lines.

Line a). — The Swedish State Railways use longitudinal equalisers alone; this is the case for this 1 C 1 (series D) locomotive with rod drive. The suspended weight is carried by the vehicle in two planes, and consequently on four points.

Line b). — Each of the two carrying bogies of the 2 C 2 locomotives of the German State Railways with rod drive corresponds to characteristic Cv, i. e. the first and the last pairs of wheels of the locomotive have each a transverse equaliser which has the effect that the load transmitted to the bogie is carried on three points, one of which is situated at the centre of the carrying axle which leads during forward running and imposes equal static pressure to the left wheel and to the right wheel on this axle.

TABLE 4.

Method of support of the 14 high-speed electric locomotives.

	1	2	3	4	5	6	7	8
	Units.	Wheel arrangement.	Carried in					
			2 planes.		3 planes.		4 planes.	
			on					
			3	4	4	6	6	8
			points.					
a	1	1 C 1	...	1 S	...	...	...	...
b	1	2 C 2	. .	...	...	1 DR (E 06)	...	...
c	4	1 Do 1	...	1 DR (E 16⁵)	1 DR (E 17) 1 NI	...	1 D R (E 16)	...
d	3	2 Do 1 and 2 D 1	3 DR (E 21, E 21⁵ and E 50)	...	...	. .	...	. .
e	1	1 Bo-Bo 1	...	1 DR (E 18)	...	...	...	...
f	1	2 BB 2	...	...	...	...	...	1 DR (E 52)
g	1	Co-Co	...	1 SpN	. .	...	...	...
h	1	1 Co-Co 1	...	1 SpN	...	...	...	...
i	1	2 Co-Co 2	...	1 S_pN	...	...	...	...
Total	14	units	3 (21.4 %)	6 (42.8 %)	2 (14.3 %)	1 (7.17 %)	1 (7.17 %)	1 unit (7.17 %)

In this locomotive, as in all other German electric locomotives with leading carrying bogies, the main frame is supported on a transverse knife edge on each of the bogies. The centres are constructed like those of the bogies of the 4-6-2 steam locomotives, table 3, column H, line B). The driving pairs of wheels form a separate group of support. The sprung weight is therefore supported at six points.

We will explain in chapter III how this method of support acts on an irregular surface of the track.

Line c.) — Among the four types of 1 Do 1 locomotives, there are three (series E 17 and E 16 of the German State Railways and one locomotive of the Dutch East Indies State Railways) which are fitted at the ends with bogies consisting of a pair of carrying wheels and a pair of guiding wheels. The fourth type, the E 16⁵ of the German State Railways, has independent truck wheels.

In the E 17 locomotives, the bogies at the two ends are like the Krauss-Helmholtz type; they are built on the Kleinow system (Allgemeine Elektrizitätsgesellschaft, Berlin) because the Westinghouse quill drive takes up all the room on the body of the axle and makes it necessary to use outside frames. Figure 13 shows

this bogie. The driving pair of wheels has 15 mm. (19/32 inch) side play in the boxes each way; the axle boxes themselves have no side play in the main frame of the locomotive. Consequently the side movement of the main frame on this pair of wheels takes place under the action of the directing force applied at the bogie centre; the driving pair of wheels which by the friction between the wheel and the rail, opposes the lateral displacement, runs until its outer wheel touches the outside rail of the curve; during this time, the axle boxes bearings move more easily in the transverse direction on the smooth and well lubricated crank pins than the wheel on the rail. For details regarding the guiding by this type of bogie, see Chapter IV.

At the front end, the pony truck is spring supported on the carrying pair of wheels; at the back end it is linked to the main frame. This latter is carried on the four driving pairs of wheels connected by longitudinal equalisers, in one plane, i. e. on two points; it is also supported by the carrying pairs of wheels at their transverse centre, where a slide with hemispherical bearings absorbs the pressure of the centres of the main frame. Consequently the main frame is supported in three transverse planes on four points (lozenge.)

The 1 Do 1 locomotives of the Dutch East Indies State Railways, (1.067 m. [3 ft. 6 in.] gauge) with individual axle drive on the Buchli system, have bogies with a carrying and a driving axle of the Buchli type with 2 m. (6 ft. 6 3/4 in.) wheel base only, which because used for the first time on these locomotives, is also known as the Java bogie. Figure 14 shows this method of construction. The carrying pair and the driving pair of wheels are assembled in a bogie in the frame of which the two axles are rigidly mounted. The pivot is not placed between these two axles but outside them, close to the driving pair of wheels. As the driving pair of wheels can turn relatively to the main frame the striking angle of the driving wheels on curves is made relatively less than on the Krauss-Helmholtz bogies.

The four driving pairs of wheels are connected by longitudinal equalising gear into a single group of supports; the load is applied to the carrying wheels, as shown in the figure, at their cross centre, by a link, the spherical end of which can slide on the carrying axle 70 mm. (2 3/4 inches) on each side. The main frame of the locomotive is therefore supported in three transverse planes on four points. Pairs of laminated springs mounted above the middle of the carrying pair of wheels, with an initial tension of 1 300 kgr. (2 860 lb.) and a final tension of 2 900 kgr. (6 380 lb.) maintain the bogies central and counteract hunting movements.

The 1 Do 1, E 16 locomotives of the German State Railways with individual axle drive on the Buchli system, are fitted with Krauss-Helmholtz bogies at both ends. Five of these locomotives had originally bogies of the Buchli type described above. The essential difference between these two types of bogie consists first of all in that the driving pair of wheels of the Krauss-Helmholtz bogie is mounted in the main frame itself, whereas in the Java bogie it is carried together with the carrying pair of wheels in a special frame; consequently the axis of rotation of the Krauss-Helmholtz bogie is situated between the pairs of wheels and that of the Java bogie behind the driving pair of wheels. At the moment these locomotives were purchased, those concerned had not completely elucidated the question of knowing if the Buchli type would be more satisfactory for these locomotives (cf. *Elektrische Bahnen*, March 1927, pp. 73 to 94). It was therefore decided to build five locomotives with bogies of one type and five locomotives with bogies of the other type.

The comparative tests and the results obtained in service show that from the

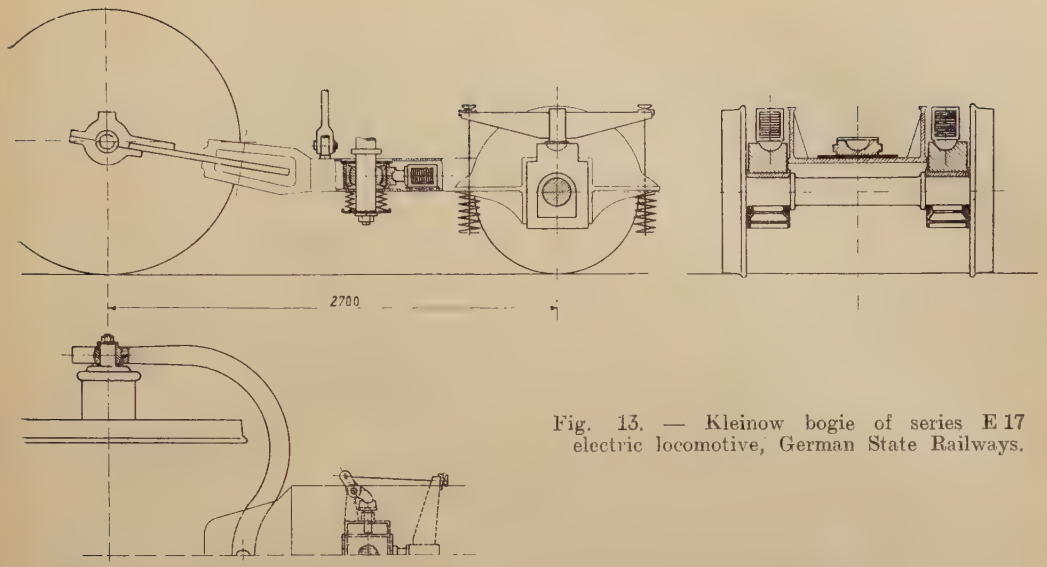


Fig. 13. — Kleinow bogie of series E 17 electric locomotive, German State Railways.

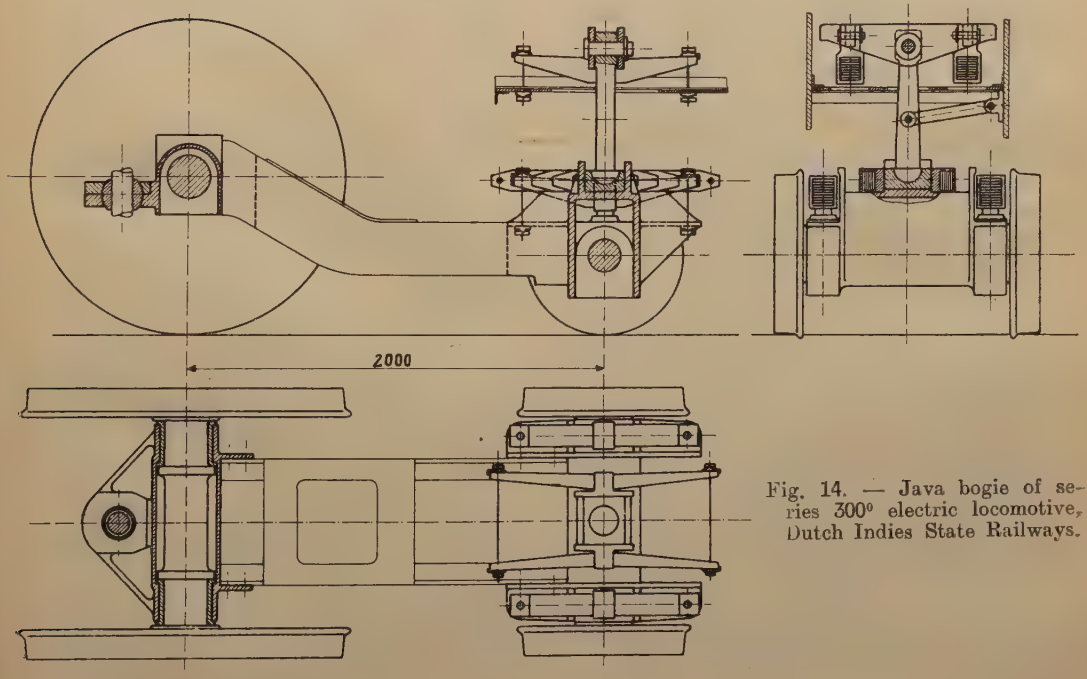


Fig. 14. — Java bogie of series 300⁰ electric locomotive, Dutch Indies State Railways.

point of view of smooth running at speeds of over 80 km. (50 miles) per hour, the locomotives fitted with the Krauss-Helmholtz bogies behaved better than those fitted with the Buchli bogies. The article mentioned above gives more ample details on this subject. The Buchli bogies were then altered into Krauss-Helmholtz bogies. The change was easily made, as these two types of bogies had been designed in view of the ultimate substitution of one or the other. In the two types of bogies the centring device consisted of a single laminated spring which acted on the centre of the Krauss-Helmholtz bogie, while on the Buchli bogie, although mounted in the same way in the main frame, it was only a stop on the spring placed in the bogie.

Against these inferior results obtained with the Java bogies, it is desirable to mention those obtained with the 1 Do 1 locomotive mentioned above, of the Dutch East Indies State Railways. According to an article published by an engineer, Mr. Hug, in « *Elektrische Bahnen* », November 1928, p. 341, the running qualities of these locomotives on the straight and on curves are excellent. On the straight [1.067 m. (3 ft. 6 in.)] gauge the locomotives reach a speed of 105 km. (65.2 miles) per hour. In view of the divergency of opinion on this bogie, we are led to think that the details of construction have played a large part in the working of these bogies. We again deal with this question in Chapter IV when speaking of the guiding of the vehicle by the bogies.

The carrying wheels of the 1 Do 1, series E 16, locomotive of the German State Railways are not linked up as regards load with the nearest pairs of driving wheels; they are loaded at their transverse centre. The 4 pairs of driving wheels are connected together by longitudinal equalisers. The sprung weight of the locomotive is therefore supported in four planes on six points.

The 1 Do 1, series E 16^5, locomotive of the German State Railways, driven by axle-hung motors has articulated pony trucks connected with the two neighbouring pairs of driving wheels by longitudinal equalisers. The suspended weight is therefore supported in two planes on four points.

Line d). — The two German 2 Do 1 locomotives, series E 21 with Westinghouse drive and series E 21^5 with Linke-Hofmann-Lauchhammer drive, have four-wheeled carrying bogies of characteristic *Cv*; the older German locomotive, 2 D 1, series E 50, with rod drive, has a bogie with two carrying pairs of wheels of the characteristic *Ce*. Above the pony truck wheels of the three types is placed a transverse equaliser. The four pairs of driving wheels and the pony truck wheels of these three types are connected together by longitudinal equalisers. As we have noted several times, since the carrying bogies of the standard German locomotives are loaded on a transverse knife edge, the suspended weight of the locomotive is supported in two planes on three points.

Line e). — The German 1 Bo-Bo 1 locomotive, series E 18, with axle-hung motors has two motor bogies, each with two pairs of driving wheels and one pair of pony truck wheels which, as regards load, is connected with the first driving pair of wheels by longitudinal equalisers. The method of drive involves the use of outside frames. The body of the vehicle is carried on these two motor bogies, connected together by a central articulated coupling, so as to insure each motor bogie being loaded on a transverse plane at three elastic supports which rest on slides. The body of the vehicle is connected with each bogie by a centre pin. One of these centres has longitudinal play on the motor bogie to enable the locomotive to run freely on curves.

The suspended weight is therefore carried by each motor bogie in two transverse planes on four points.

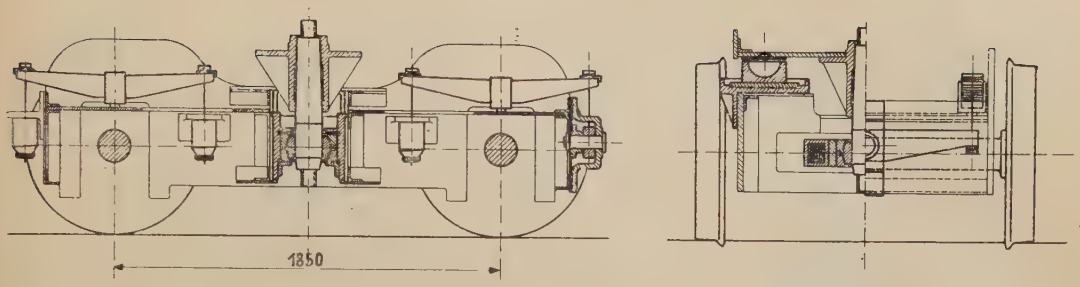


Fig. 15. — Bogie of series E 52 electric locomotive, German State Railways.

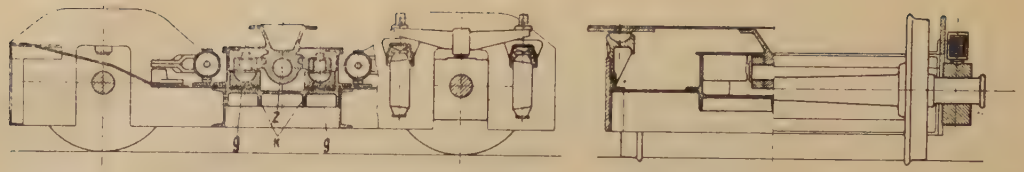
Line f). — The bogies of the German 2 B B 2 locomotive, series E 52 with rod drive, correspond to the characteristic C*v*. Figure 15 shows the method of construction. The driving pairs of wheels are connected in pairs by longitudinal equalisers. The suspended weight is therefore supported in four transverse planes on eight points.

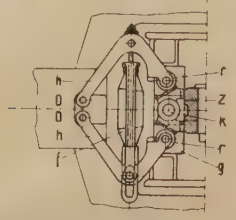
Lines g) *and* h). — On the Co-Co locomotive, series 7000, of the North of Spain Railways, driven by axle-hung motors, the body of the vehicle is supported on each of the two motor bogies by a sort of bolster, which has however no lateral play. These cross girders rest on each motor bogie on a pair of elliptical springs arranged on each side of the longitudinal centre line of the vehicle, the springs being like the bolster springs of carriage bogies. The weight of the body is consequently supported in two transverse planes on four points. Each bolster carries, like on bogie carriages, a centre which in this case is of course much larger, and is fitted in addition on both sides of the centre line of the vehicle with auxiliary elastic supports, principally intended to prevent any transverse inclination of the body.

The motor bogies themselves are each one supported in two transverse planes on four points. The longitudinal equalisers are placed between the two inner driving pairs of wheels of each bogie.

The second locomotive mentioned in the reply from the North of Spain Railways, the 1 Co-Co 1, series 7100, is of exactly the same construction as the series 7000 locomotive described above. The pairs of wheels of the leading pony truck are of the Adam type; a cylindrical helical spring, 561 mm. (1 ft. 10 5/64 in.) long placed between the frames, acts as a centring device, i. e. as a method of making the truck take part in the guiding. On this locomotive also, the suspended weight is supported in two planes on four points; the bogies are supported each on four points as in the preceding case.

Line i). — The 2 Co-Co 2, series 7200 locomotive of the North of Spain Railways, with Buchli drive, has carrying bogies, the details of which deserve particular attention as regards question III (fig. 16). Actually in this method of construction as developed by the Winterthur Swiss Locomotive and Machine Works, the springs of the centring device in addition have to counteract the hunting movements of the bogies. Special arrangements are provided to meet this condition. On the other hand this method of construction should not increase to any extent the pressure of the flanges of the guiding wheels of the bogies, pressure which in the usual types of centring device increases directly as the lateral movement of the centres. Figure 16 shows this bogie and





Position : Longitudinal centre line of bogie under longitudinal centre line of locomotive.

Fig. 16. — Bogie of series 7200 electric locomotive, North of Spain Railways.

its details. We will return to a consideration of these latter in Chapter IV.

By its method of support the bogie corresponds to the characteristic Cme.

The three pairs of driving wheels of each driving bogie are connected together by equalisers into a group support, so that each driving bogie has three points of support.

Half-way between the first and the second pairs of driving wheels is the centre for the whole of the motor bogie. The body of the locomotive here rests on a hemispherical carrying centre. Transverse oscillations are prevented by hemispherical side friction blocks kept in contact by conical springs. Body stops are used to prevent the superstructure from lifting off the motor bogie. The body can therefore be considered as supported on four points.

The following may be said to sum up the position as regards electric locomotives :

Out of the 14 electric locomotives mentioned in the replies, 3 (21.4 %) are carried on three points, 8 (57.1 %) on four points, 2 (14.3 %) on six points, and 1 (7.2 %) on eight points. 64.2 % of the locomotives are supported in two planes, 21.47 % in three planes, and 14.34 % in four planes.

Among these 14 electric locomotives, 3 (21.4 %) have bogies with two pairs of carrying wheels at both ends, and 3 (21.4 %) at one end; out of these 6 bogies, 4 (66.6 %) correspond to the characteristic Cv and 1 (16.7 %) to the characteristics Ce and Cme.

The 3 locomotives fitted with bogies for one direction of running have pony trucks for the other direction of running.

Among the 14 — 6 = 8 remaining locomotives, 3 (21.4 %) have bogies consisting of one carrying and one driving pair of wheels, 4 (28.6 %) have a pony truck at each end, 1 locomotive (7.2 %) has no carrying pairs of wheels, the guiding being done by the six-wheeled driving bogies.

Tenders.

As far as concerns tenders regarded as vehicles, some of the Administrations have not given any information, and that sent by others is incomplete. The eight-wheeled tender is the usual type. The use of four pairs of wheels has been made necessary primarily because of the large quantities of water and fuel that must be carried.

The Netherlands Railways and also, as we learnt shortly before completing this report, the Danish State Railways,

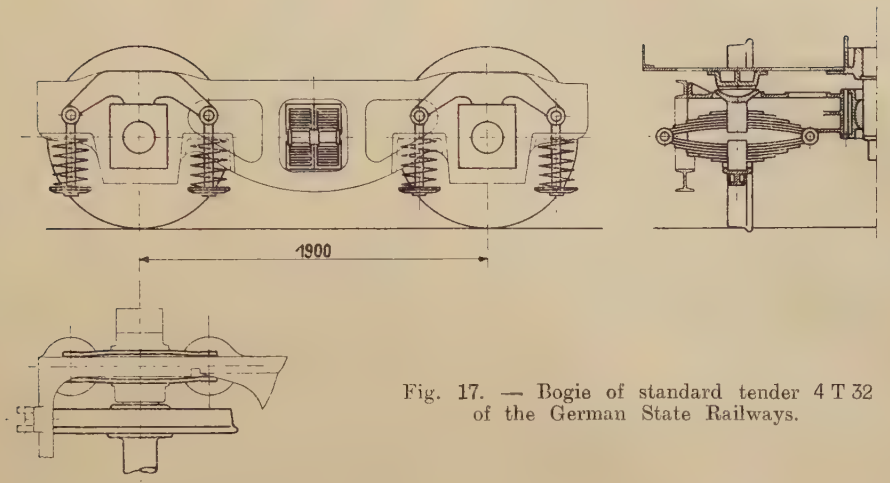


Fig. 17. — Bogie of standard tender 4 T 32 of the German State Railways.

carry the four pairs of wheels of the tenders of their 4-6-0 (series 3900 and series R) locomotives in a common frame; all the other Railway Administrations group the four pairs of wheels in pairs in two bogies or fit two pairs of wheels in a leading bogie and the other two pairs of wheels in the main frame of the tender.

It will be easier to appreciate the more or less satisfactory properties of the different methods of construction if we consider rapidly the most widely used types.

The standard German locomotive tender for main lines, which is probably the most modern of the tenders mentioned in the replies, is carried on two four-wheeled bogies with double suspension. Figure 17 shows this method of construction. The centres of the bogies are fixed in two-piece centre castings but do not carry any load. Two bearings are fitted in the same plane on each side of the bogie centres. Previously, as on bogie carriages, bolsters with lateral play were used; this arrangement has been given up. The flat friction blocks in cast steel are bolted beneath the body cross bearers. Each rests on a spherical cast iron roller, with a large bearing surface, which can follow any inclination of the bogie by adapting itself to the concave surface of the bracket taking the transverse springs (two pairs of elliptical springs). On the lower face of the bracket are inserted distance pieces which engage with the buckles of the two side by side transverse springs and prevent them from moving relatively to the bearing surface.

The bogie frame built up of steel castings is carried over each axle box by two helical springs made from round sections. These springs are held in pockets cast with the frame guides. The load is transmitted in line with the frames carrying the body of the tender to the axle boxes, the helical springs of which are placed in line with the guides. The two helical springs [strength = 105 kgr. per mm. (5 880 lb. per inch)] of each box transmit the tractive effort to the box by the spring bolts and two flat stirrups on each side of the guides.

The boxes allow each pair of wheels of the bogie to incline 40′ transversely to the bogie frame, so that a wheel on

one side can rise or fall 17.4 mm. (11/16 inch). In the axle boxes themselves there is a bearing plate, the top face of which is convex in order to transmit the load without shock. The method of support corresponds to characteristic C*e*.

In the same way the tenders of the 4-6-2 series 3000, 4-8-0 series 4000, and 4-8-2 series 4600 locomotives of the North of Spain Railways have bogies with bilateral supports on spherical rollers, fixed centres, but single spring gear, consisting of four laminated springs. The method of support corresponds to the characteristic C*e*.

Many diamond frame bogies of the American type with the side frames built up of flat bars to which the four axle boxes are rigidly bolted are still in use. The tender body is carried in these bogies either by means of two rollers on a hemispherical flat bearing, each of which rests on two pairs of elliptical springs, or by means of a disc in the usual position of the centre. As we have already explained, a bearing on a central disc is not equal to the « centre knife edge ».

Bogies of this type could not possibly be used at the leading end of a locomotive because their wheels cannot adapt themselves to inequalities in the rail surface without bending the side frames. This is an undoubted defect in this type of bogie. It is only on account of their short wheel base that this defect has not made itself more obvious. Diamond frame bogies of this type have been reported with a wheel base of 1.80 m. (5 ft. 10 55/64 in.) (German State Railways, Finnish State Railways, and North of Spain Railways) and 1.90 m. (6 ft. 2 5/32 in.) wheel base (Dutch East Indies State Railways, Bergslagen Railway).

The Madrid-Saragossa-Alicante Railways carry the tenders of their 4-6-2 locomotives on two four-wheeled bogies. The load is transmitted to centre discs with pins and therefrom to longitudinal equalising springs suspended on drop equalisers. No other railway uses tenders with longitudinal equalising springs, i. e. a method of construction quite usual in the case of the leading bogies of locomotives and moreover a very practical one, but which in this case as on carriages involves the use of outside boxes. In Chapter III we show the effects of outside springs, compared with inside springs. The method of support of these bogies comes under the characteristic C*m's*.

Eight-wheeled tenders with a single bogie have been built by the German State Railways as a transition type, in order to be able to turn standard 4-6-2 locomotives with their tenders on 20-m. (65 ft. 7 in.) diameter turntables, or with the hope of obtaining particularly good running. The leading bogie is supported on four points. The body of the tender rests on the bogie, in one group of tenders, by the arrangement with cross knife edge, and in the other group, by a spherical centre. The methods of support therefore correspond to the types C*e* and C*m'e*. The two trailing pairs of wheels carried in the tender frame are connected by equalisers. Consequently the tender itself is supported on either four or three points.

This type of vehicle with two pairs of wheels in a bogie and two pairs of wheels rigidly carried in the main frame makes it possible to use short tenders; but from the point of view of our investigation it cannot be recommended, as two independent bogies (diamond frame bogies must be eliminated for similar reasons) adapt themselves better to irregularities in the surface of the track and consequently are subject to smaller changes of pressure of the guiding wheels. Another argument is that unless the flanges of the wheels following the bogie are reduced by 10 to 15 mm. (25/64 to 19/32 inch), i. e. the first pairs of wheels mounted rigidly in the frame, it is not possible for the tender to run through turnouts of small radius.

The result of the above is that the eight-wheeled tenders of the locomotives dealt with in the replies are carried in one case only on four pairs of wheels in a common frame, and in every other case on two bogies with two pairs of wheels or one bogie with two pairs of wheels and two pairs of wheels rigidly mounted in the tender frame. The type carried on two bogies is to be preferred.

The bogies correspond, by their method of support, to the characteristics Ce *and* Cm*'s, unless it is a question of bogies with pairs of wheels rigidly mounted in the bogie frame, in which case we have not adopted any characteristic, because considered from the point of view of support, they are statically indeterminate and should not therefore in our opinion be used under any future tenders.*

In general the wheels of the guiding bogies of steam locomotives carry smaller loads than driving and coupled wheels.

On the standard German 4-6-2 type locomotives the reduction is 18 and 20.6 %; in the case of the locomotives of the former German States, it lies between 15.7 and 29.8 %. The Finnish State Railways report a reduction in load of 27.8 % on the wheels of the leading bogies of their H 8 and H 9, 4-6-0 type locomotives; they say that the results have been satisfactory and that they adopted these reduced loads to prevent derailment. The reduction is 30 % on the 4-8-2, 39.3 % on the 4-6-2, 42 % on the 4-8-0 and 43.8 % on the 4-8-4 Spanish locomotives.

The load on the wheels of the leading trucks of steam locomotives is as much as 21.8 % less than on the following pair of coupled wheels.

The reduction of the load on the leading bogies is therefore very great and the evident reason is that it has been desired to avoid excessive surface pressures which would result in the rapid wear of these small wheels which in part are submitted to considerable braking forces.

From Hertz's equation relating to surface pressure (a roller on a smooth plate, *Hütte pocket-book*, 26th edition, Vol. 1, p. 683), the pressure is among other things inversely proportional to the square root of the radius of the carrying wheel. Thus a wheel of 1600 mm. (5 ft. 3 in.) radius will affect the pressure by 1/40 and a wheel of 400 mm. (1 ft. 3 3/4 in.) radius by 1/20.

The driving and coupled wheels of one and the same steam locomotive are equally loaded. No Administration has considered it necessary to reduce the load on the leading coupled pair of wheels as compared with that on the following coupled pairs.

The trailing carrying wheels have generally wheels of 1.25 m. (4 ft. 1 13/64 in.) diameter and are usually less heavily loaded than the coupled wheels.

On the electric locomotives also, the weight on the different pairs of driving wheels are equal to each other, even though the driving pair of wheels is also guiding, as for example the driving bogies of the Co-Co, series 7000, locomotives of the North of Spain Railways, or if it occupies an intermediate position.

Against this, the pairs of carrying wheels are always less heavily loaded than the corresponding driving pairs. This difference is for example 30 % for the 2 C 2, series E 06 locomotives of the German State Railways; for the whole of the electric locomotives it is on the average 21.8 %.

The *unsprung weights* of the electric locomotives are less than those of the steam locomotives; actually the wheels are smaller in diameter, and with the individual drive of the wheels the weight of the rods and the counterbalance weight is saved. The greatest unsprung weight is found with the axle-hung motor drive (for example 4.55 tons on the E 18 locomotive of the German State Railways) in which, as is known, half the weight of the motor is unsprung.

All the Administrations consulted have endeavoured to so arrange the water and

coal on the tender that the axle weights change relatively to one another as little as possible as the fuel and water are used. By careful weighing, several Administrations have satisfied themselves that this is so. The ideal solution which consists of making the common centre of gravity of the water and fuel coincide with the centre of gravity of the suspension comes up against practical difficulties, especially owing to the form of the coal hopper which has to be so designed as to make sure that the fuel feeds itself forward as it is used.

The good running of a steam locomotive depends very largely on the height of the centre of gravity, as the elevated position of the centre of gravity is favourable to smooth running. The height of the centre of gravity increases with the increase in heating surface, i.e. inevitably, because through this ever-increasing size, the centre lines of the boilers have constantly to be raised. Probably with the centre line of the boiler 3.40 m. (11 ft. 1 55/64 in.) over the rail level (2-8-4 locomotive of the Austrian Federal Railways) the limit with the actual loading gauge has been reached. We are not speaking here of special types, such as the ultra-high-pressure Schwartzkopff-Löffler locomotive, the centre line of the boiler of which is 3.63 m. (11 ft. 11 in.) above rail level. As the overturning moment which affects the stability on curves increases with the square of the speed and directly with the height of the centre of gravity, above the rail level, it is necessary when dealing with the question of stability to know the height of the centre of gravity. Unfortunately the positions of the centres of gravity of all the locomotives mentioned in the replies are not known. From the information supplied the height of the centre of gravity of the high-speed locomotives above rail level varies between 1.64 m. (5 ft. 4 1/2 in.) (4-6-0 Finnish locomotive) and 2.235 m. (7 ft. 4in.) (4-8-2 locomotive of the North of Spain).

The height of the centre of gravity of the standard gauge locomotives is given approximately by the empirical formula:

$H_s = 1.034\ H_k - 1.20$ (dimensions in metres)

in which H_s = the height of the centre of gravity of the locomotive as a whole above rail level;

H_k = the height of the centre line of the boiler above rail level.

For example :

— For the standard German locomotives, series 01 (bar frames),

$H_s = 1.034 \times 3.100 - 1.200 = 2.005$ m. (6 ft. 3 25/64 in.) against the actual 2.005 m.;

— For the standard German locomotive series 03 (bar frames),

$H_s = 1.034 \times 3.100 - 1.200 = 2.005$ m. (6 ft. 3 25/64 in.) as compared with the actual 2.008 m. (6 ft. 3 33/64 in.);

— For the former Baden locomotive 4-6-2 series 18^3 (bar frames),

$H_s = 1.034 \times 2.900 - 1.200 = 1.799$ m. (5 ft. 10 53/64 in.) as compared with the actual 1.780 m. (5 ft. 10 5/64 in.);

— For the 4-6-0 Finnish locomotive (plate frame),

$H_s = 1.034 \times 2.725 - 1.200 = 1.618$ m. (5 ft. 3 45/64 in.) as compared with the actual 1.640 m. (5 ft. 4 35/64 in.);

— For the former Bavarian 4-6-2 locomotive series 18^5 (bar frame),

$H_s = 1.034 \times 2.855 - 1.200 = 1.752$ m. (5 ft. 9 in.) as compared with the actual 1.740 m. (5 ft. 8 35/64 in.);

— For the 4-6-2 Swedish series F locomotive (plate frame),

$H_s = 1.034 \times 2.900 - 1.200 = 1.799$ m. (5 ft. 10 53/64 in.) as compared with the actual 1.850 m. (6 ft. 53/64 in.).

The centre of gravity of the spring carried weight of steam locomotives is between 1.90 m. and 2.32 m. (6 ft. 2 25/32 in. and 7 ft. 7 23/64 in.) above

rail level. For the 4 locomotives with bar frames (series 01, 03, 18^3 and 18^5 of the German State Railways all of the 4-6-2 type) the height of the centre of gravity H_{sg} of the spring carried masses above rail level has been calculated. *The empirical formula :*

$H_{sg} = 1.368\ H_k - 1{,}935$ (dimensions in metres) *gives approximate agreement with the calculated figures.* For example we find :

— For the standard German locomotive series 01 and 03 (bar frame),

$H_{sg} = 1.368 \times 3.1 - 1.935 = 2.305$ m. (7 ft. 6 3/4 in.) compared with the actual 2.290 m. (7 ft. 6 5/32 in.) for 01, and 2.320 m. (7 ft. 7 23/64 in.) for 03, for the former Baden 4-6-2 locomotive series 18^3 (bar frame),

— $H_s = 1.368 \times 2.9 - 1.935 = 2.032$ m. (6 ft. 8 in.) as compared with the actual 2.020 m. (6 ft. 7 33/64 in.) and compared with 2.020 m. of the 4-6-0 locomotive of the Berslagen Railway (bar frame), as well as compared with 1.965 m. (6 ft. 5 23/64 in.) of the former Wurtemberg 4-6-2 locomotive, series 18^1 (plate frame).

The Administrations who replied to the question state that the running of these locomotives is smooth on good track. The Dutch East Indies State Railways [1.067 m. (3 ft. 6 in.) gauge] are less satisfied with the running of the four-cylinder compound 4-6-2, series 24 locomotives, the centre line of the boiler of which is 2.30 m. (7 ft. 6 35/64 in.) above the rails, than with that of the 4-6-2, series 22 locomotive having the centre line of the boiler 100 mm. (3 15/16 inches) lower down. In addition, another administration stresses the fact that its 2-8-2 locomotive runs better than its 4-6-2, although the height of the centre line of the boiler is the same [2.90 m. (9 ft. 6 13/64 in.)] in the two cases.

The position of the *centre of gravity of the tenders* has been given in a few cases only; the dimensions quoted for H_s vary between 1.66 m. and 1.79 m. (5 ft. 5 23/64 in. and 5 ft. 10 15/32 in.), and for H_{sg} between 1.83 m. and 2.02 m. (6 ft. and 6 ft. 7 33/64 in.). For example, for the same 8-wheeled tender of the old Bavarian 4-6-2, series 18^5 locomotive, and the old Baden 4-6-2, series 18^3 locomotive, the following values have been found :

$H_s = 1.79$ and 1.78 m. (5 ft. 10 15/32 in. and 5 ft. 10 5/64 in.) and

$H_{sg} = 1.94$ and 1.944 m. (6 ft. 4 25/64 in. and 6 ft. 4 35/64 in.).

The last question of this chapter which deals with *the stability of the vehicles* is one of the principal questions in our programme. It is necessary that the vehicle should not turn over when running onto or off a curve. It is clear that besides the height of the centre of gravity of the whole and the centre of gravity of the spring carried parts of the vehicle, the superelevation of the outer rail is of capital importance. The operating department must also decide the maximum speed and the permanent way engineer take the necessary steps to guarantee stability with the permanent way in use. The rolling stock is necessarily designed on the bases of the conditions under which it has to be used. It is not in the power of the carriage builder to modify appreciably the height of the centre of gravity of a type. This remark applies particularly to the steam locomotives, on which, as we have seen, the centre of gravity has become higher and higher as the power of the boiler has increased and in which this progressive increase in height has had the favourable effect of making the running on the straight smoother and consequently safer.

There is no need to consider carriages here, on account of the low position of the centres of gravity.

Very thorough investigations have been made over many years into the superelevation of the outer line of rails.

The investigations made in 1930 by

the German Railway Association were based on the theoretical formula :

$$h_{mm} = 11.8 \cdot \frac{V^2}{r}$$

(in which h_{mm} = the superelevation, V = the speed in km./h., and r = the radius in metres of the curve), and an investigation was made into the coefficients to be adopted according to whether :

1. the wear of the rail in height,
2. the safety of working,
3. smooth running, and
4. the economic question (the wear of the rails in height and width) are considered as the preponderant factors. The result of these investigations has been condensed into paragraph 20, section 2) of the « Technical Standards ». An important point is that from the point of view of the construction and maintenance of the track, it is not desirable that the centre of gravity of the locomotive as a whole should exceed 2.10 m. (6 ft. 10 43/64 in.) above rail level, whereas the Austrian locomotives 2-8-4 types 114 and 214 with the centre line of the boiler 3.40 m. (11 ft. 1 55/64 in.) above the top of the rail, have the centre of gravity at a height of 2.07 m. (6 ft. 9 33/64 in.) as otherwise the superelevation will have such a value that with short transition curves, it would be impossible to make the superelevation incline sufficiently regular, and this would have the invetable consequence of reducing the speed in such cases; such cases are frequent on certain lines. The German Railway Association recommends that the maximum superelevation should =

$$a \frac{V^2}{r} + b_1$$

and the minimum :

$$a \frac{V^2}{r} - b_2$$

The superelevation must not exceed 150 mm. (5 23/32 inches) nor be less than

$$11.8 \frac{V^2}{r} - 90,$$

this latter formula being intended to give the minimum superelevation for sufficient stability.

In the above formulæ :

h is the superelevation in mm.

V the maximum speed on the superelevated line in km./h.

r the radius of the curve in metres.

$a = 8$.

$b_1 = 30$, in exceptional cases up to 40 at the most.

$b_2 = 30$.

In addition it is recommended that the superelevation should be as high as possible within the limits laid down.

Obviously it is necessary to insist that the superelevation satisfy the condition of a sufficient margin of safety, even in the most unfavourable cases. The more it can be increased, the better the desiderata relative to smooth running of express trains, uniform wear on the top of the rails, and economy can be realised.

A question still remaining open is that of knowing what degree of safety it is desirable to insist upon. During the discussions which took place at the meetings of the technical committee on the permanent way of the German Railway Association it was thought desirable to insist upon a factor of safety of at least 5 against turning over. It is desirable therefore to see what answers the different administrations have sent; they are unfortunately very few, because they themselves do not know the position of the centre of gravity of more than a few vehicles.

The German State Railways base their calculation on a curve of 250 m. (12 ½ chains) radius. This curve was select-

ed because the factor determining the stability $\frac{V^2}{r}$ reaches in the range between $r = 500$ m. (25 chains) and $r = 180$ m. (9 chains) its maximum value when $r = 250$ m. (12 1/2 chains). The maximum speed on this curve is 60 km. (37.3 miles) per hour, the superelevation of the outer line of rails is 115 mm. (4 33/64 inches). The degree of safety has been arrived at by means of the formula given below in dealing with electric locomotives. It was found that for the two express 4-6-2 series 01 and 03 locomotives, the position of the centre of gravity of which is known, the stability $n = 6.4$ under the most unfavourable conditions.

For the former Bavarian 4-6-2, series 18^5 locomotive, we find $n = 13.1$ with $V = 60$ km. (37.3 miles) per hour, $r = 292$ m. (14 1/2 chains) and $h = 98$ mm. (3 55/64 inches); for the corresponding eight-wheeled tender $n = 12.6$.

For the former Baden 4-6-2, series 18^3 locomotive, we get $n = 10.2$ with $V = 66$ km. (41 miles) per hour, $r = 310$ m. (15 1/2 chains) and $h = 110$ mm. (4 11/32 inches).

For

$$h_{min} = \frac{V^2}{r} - 90 = 11.8 \cdot \frac{66^2}{310} - 90 = 76\,\text{mm}.$$

(2 63/64 inches), we find $n = 6.3$. A broken spring does not reduce the value of n.

Some administrations, for example the North of Spain Railways, keep the long equalising levers in place in the event of a broken spring by spring checks limiting the distance they can move.

We have examined all the steam locomotives upon which we have received, or about which we have collected, information as regards stability on a curve of 300 m. (15 chains) radius, using all the values of h communicated, corresponding to this curve, for the superelevation of the outside rail and the maximum speeds allowed. The stability which we designate by n has been calculated by means of the following considerations :

The centrifugal force

$$C = \frac{G}{g} \cdot \frac{v^2}{r}$$

counteracted by the centre of gravity of the locomotive as a whole by the horizontal transverse force of the axle load $G \frac{h}{s} \cos \alpha$ (fig. 18). The angle α being very small, we can make $\cos \alpha = 1$. Consequently the force applied at S is the excess of the centrifugal force

$$\Delta C = \frac{G}{g} \cdot \frac{v^2}{r} - G \frac{h}{s} = G \left(\frac{V^2}{127.5 \cdot r} - \frac{h}{s} \right)$$

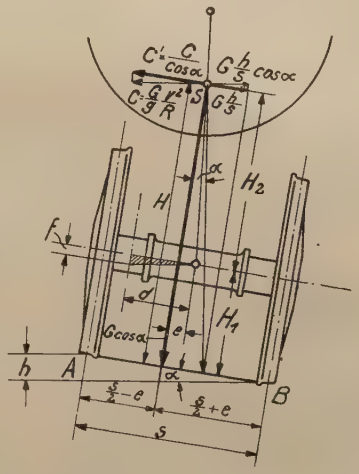


Fig. 18. — Stability.

For the degree of safety n, we have the equation :

$$n \cdot \Delta C \cdot H = G \cdot \left(\frac{s}{2} - e \right),$$

in which $e = f \cdot \frac{H_2}{d}$.

We then get

$$n = \frac{G \left(\frac{s}{2} - e \right)}{\Delta C \cdot H} = \frac{G \left(\frac{s}{2} - e \right)}{G \left(\frac{V^2}{127.5 \cdot r} - \frac{h}{s} \right) \cdot H}$$

If we take $s = 1.50$ m. (4 ft. 11 in.) we find after having made the necessary calculations the degree of stability :

$$n = \frac{127.5\,(0.75 - \epsilon)\,r}{H\,(V^2 - 85 \cdot h \cdot r)}$$

These purely « static » considerations have been given new additions of very great value, by the Technical Committee of the German Railway Association, as the result of the investigations made by the Austrian Federal Railways on the dynamic actions which occur after the fracture of a spring. The two methods end in the same result, so that it has been thought reasonable to use for h the formulæ reproduced above.

The calculations have brought out the following facts :

The stability of the German express steam locomotives, when running at 65 km. (40.4 miles) an hour on a superelevation of 110 mm. (4 11/32 inches) and a maximum deflection of the springs of 50 mm. (1 31/32 inches) lies between 8.5 (series 01) and 10.1 (series 18^5 Bavarian), i. e. between quite narrow limits.

The 4-6-0 locomotive of the Norwegian State Railways under almost identical conditions has a factor of stability of 10.3.

The Netherlands Railways' locomotives which run through a curve of 300 m. (15 chains) radius at a maximum speed of only 60 km. (67.3 inches) per hour, have a factor of safety of 15.5 to 19.5.

The degree of stability of the 4-6-0 Danish locomotive is slightly less than that of the German Steam locomotives; it equals 7.4, on a curve of 250 m. (12 1/2 chains) radius, it is true run over with 95 mm. (3 47/64 inches) superelevation, at the maximum speed of 60 km. (37.3 miles) per hour. In the case of the curve $r = 300$ m. (15 chains), no information was supplied.

The greatest degree of stability of all the vehicles of standard gauge examined is that of the 4-6-0 locomotives of the Bergslagen Railway with the factor of 27.5; it reaches this value at a maximum speed of 60 km. (37.3 miles) per hour ont the curve of $r = 300$ m. (15 chains), thanks to a superelevation $h = 120$ mm. (4 23/32 inches) and to the low position, 1.77 m. (5 ft. 9 43/64 in.) above rail level, of the centre of gravity of the locomotive as a whole.

In order to be complete, we will also mention that the Swedish standard gauge locomotive 4-6-2 has a stability of 23.4 on a curve of r = 450 m. (22 1/2 chains), the smallest that is met on the line, V = 70 km. (43.5 miles) per hour and $h = 105$ mm. (4 1/8 inches).

The broad gauge locomotives mentioned in the replies have a factor of stability of 22.9 to 54. For example on the Finnish 4-6-0 locomotive the factor of safety equals 22.9 for 1.524-m. (5 feet) gauge at a speed of 60 km. (37.3 miles) an hour on a curve of 300-m. (15 chains) radius, with 120 mm. (4 23/32 inches) superelevation. With the 1.67-m. (5 ft. 5 3/4 in.) gauge locomotives of the North of Spain Railways the stability is as much as 48 to 54 when running through a curve of 300 m. (15 chains) radius with 150 mm. (5 29/32 inches) superelevation at 60 km. (37.3 miles) per hour.

As regards electric locomotives we have received detailed information from the German State Railways. The stability of the worst locomotives with single motor above the frame is, from tests carried out, $n = 7$. The locomotives with individual axle drive owing to the much lower position of their centres of gravity have a higher degree of safety against overturning.

We find therefore quite definitely that the satisfactory solution of the question of stability just considered presents no difficulty, and that it is possible to provide a sufficient degree of stability.

CHAPTER III.

Adaptation of the running gear to surface irregularities on transition curves and to unevenness in level of the track.

In the foreword to their questionnaire, the reporters pointed out that the safety of vehicles at high speeds required the fulfilment of certain conditions relating to the construction and maintenance of the vehicle and of the track and that, as far as the vehicle was concerned, the method of transmitting the suspended weight to the wheels was one of the principal questions that awaited the consideration and appreciation of the reporters. This is the reason why it has been necessary to deal at such great length in chapter II of this report with the method of carrying the main frame of the locomotive and on the details of the running gear which are responsible for guiding the locomotive at the leading end. It is obvious that a guiding wheel on which no reduction of weight ever occurs cannot in any case be pushed off the line by the static guiding pressure Y acting transversely to the rail from the inside to the outside, provided that it is suitably loaded. The more perfectly this condition is fulfilled, the more the static surety of guiding is guaranteed, provided, of course, that the angle of contact of the directing wheel with the rail remains small and that there are no notable irregularities in the direction of the rails to set up dynamic actions. Moreover it is known that one-sided irregularities in the track on the straight and on curves, but above all alterations in height of the points of support of the wheels, alterations which arise when running off a superelevated curve onto the irregular surface of the transition, are the causes capable of altering most appreciably the static pressures, corresponding to points of support of equal height, of the leading wheels of the locomotive, i. e. of the most important pair of wheels of the whole of the wheels and to compromise the safety of running.

Tests and experience have taught us how these dangers can be guarded against. In the hard school of success and failure, satisfactory designs have been found for the running gear. But the question still remains as to what extent the different types satisfy the desired object. In order to be able to reply to this question we require some scale of comparison that we can then apply to the bogies examined in Chapter II. Actually only bogies consisting of carrying wheels or of carrying and driving wheels should be taken into consideration as regard the guiding of locomotives at high speeds. This scale then should enable us to compare, independently of any personal judgment, the numerical value of the different types of bogies. How are we to find this impartial standard?

Continuing the work of Dr. Lindemann and Professor Denecke, Dr. Uebelacker has shown in the *Organ fur die Fortschritte des Eisenbahnwesens* for the years 1926 and 1928 how the variations of the pressures of the wheels of a vehicle when running on irregular track surfaces can be calculated. By means of this new method we can find the needed scale for comparing our bogies. But our investigations have shown broadly in a manner generally applicable, the relations between the number of planes of support of the locomotives, the way of carrying the bogie and the changes of pressure of the directing wheels. We proceeded as follows :

Let us suppose a locomotive on a straight level line, the points of support of all the wheels being at the same level, except the left wheel of the first pair of wheels of the directing bogie with two pairs of carrying wheels, which is placed at X millimetres (mm.) below the horizontal plane passing through the

points of support of all the other pairs of wheels. Let δ kgr. per mm. be the strength of the springs of the pairs of wheels carried in the main frame of the locomotive, C the « type of support characteristic » frequently mentioned in Chapter II, i. e. a numerical value representing the statical quality of the method by which the bogie carries its load, and therefore also takes into account the more or less great difference between the springs of the bogie and those of the main frame. Let Δ P represent the value of the alteration of pressure of the left hand leading wheel when the leading carrying guiding pair of wheels is in the hollow of X mm., i. e. in an out of line position relatively to the other pair of wheels placed horizontally; we then have :

$$\Delta P = C\delta X.$$

The variation of pressure of the wheel remains relatively small if the two factors C and δ are kept small and if, consequently, the product C δ has a minimum value. Actually the value X of the hollow place in the track is not within the control of the rolling stock builder as it depends on the state of the line run over. We shall see later by what method it is possible to bring the characteristic C to a low value. But for the value of the variation of pressure Δ P of the wheel of the leading guiding bogie, it is no less important that the springing of the driving wheels and coupled wheels should be flexible, i. e. that δ should be relatively small.

When the locomotive is placed as above from the static point of view it is in the same position as when its second pair of bogie wheels passes from a superelevated curve down a leaving ramp, the first pair of bogie wheels then running on the slope of the leaving transition and its guiding wheel having dropped by X mm. We know that on the transition up to the superelevated rail of the curve the guiding wheel is overloaded as it is not only the highest wheel of the locomotive but it also receives an extra load through the transmission of the guiding pressure to the inside wheel and to the main frame. The inside wheel of the leading directing pair of bogie wheels can have the load taken off it to a very serious extent when through the action of the excess centrifugal forces or of shock, the main frame lifts on the bogie. But considered as a whole, the conditions relative to the changes of pressure of the directing wheel are more favourable when running onto the superelevated curve than when leaving it. For this reason, we will consider the conditions found on the leaving ramp.

We will also suppose that we are dealing with a 4-6-2 locomotive carried *in three planes* and arranged in such a way that we can fit it in turn with different types of bogies with all the various types of support considered in Chapter II. It may be recollected that the bogies on which the main frame is carried on the two sides, to the right and to the left of the transverse centre line, were given the characteristic C. In addition among these bogies there were some on which the main frame is supported at the centre of the bogie by a spherical or hemispherical centre; we have distinguished them by the characteristic C*m*, or by C*m*', if the support consisted of flat circular disc centre bearing blocks. To be exact a support of this kind becomes bilateral; but with the small diameter of the mean surface pressure circle on such a disc, it approximates to the support on a hemispherical centre. Remember too that we added to these characteristics C, C*m* or C*m*', the letters *e*, *s*, *l*, and *v*, if the bogie transmitted the load from the main frame with its own sprung weight by four sets of independent springs (*e*) to the 4 axle boxes of the two pairs of carrying wheels, or by two longitudinal springs (*s*), or by four individual sets of spring gear with lon-

gitudinal equalisers (*l*), or by three points if there was an equaliser at the leading end (*v*). Bogies with equalisers at the back represented by the characteristic C*h* are not found on any of the locomotives mentioned in the replies. But as a more careful examination has brought out the fact that for the leading left wheel considered C*h* = C*e*, we have also included, for completeness, this type of support in the diagrams of figures 19 to 21.

The value of Δ P is very much affected according as to whether the bearing springs are placed inside the sole bars or outside them, and also by the value of the ratio $\frac{f}{s}$ wherein *f* is the transverse distance apart of the springs and *s* the transverse distance apart of the bearing points of the wheels.

As we dealt with this matter in all its details elsewhere — here the necessary space is not available — these values of C have been calculated for the different methods of support and shown as ordinates above the values of the abscissæ $\frac{f}{s}$. Figure 19 shows the curves obtained in this way for a locomotive supported in three transverse planes. These curves apply to the 4-6-2 locomotive reproduced in the same figure, a locomotive which from the point of view of the position of the planes of support resembles the former Bavarian S 3/6 locomotive. Special investigations have shown, however, that these diagrams are also applicable to all 4-6-2 and 4-8-2 locomotives on which the ratio of the distance between the middle and front of the bearing planes and the distance between the planes of the trailing and leading supports is the same as on the locomotive considered, that is represented in figure 19, for which this value was 0.4179. Any other types of 4-6-2 locomotive supported in three planes have other values for C, but relatively the values C*v* of these locomotives would be statically greater than the values C*e* or C*s*, etc., in about the same proportion. For each method of support (C*e*, C*s*, C*v*, etc.) we have drawn three curves to which we give the designations $p = 0.7$, $p = 0.9$ and $p = 1.1$. These values of p give us the proportional strength of the bogie springs, relatively to that of the main bearing springs. Actually only this proportional strength of the springs is brought into the mathematical expression C and not the strength of the springs of the main frame or of the bogie. If, for example, the strength of the main bearing springs is $\delta = 100$ kgr. per mm. (5 600 lb. per inch), and if the strength of the bogie bearing springs C*e* is $e = 143$ kgr. per mm. (8 008 lb. per inch), we get

$$p = \frac{\delta}{\delta_e} = \frac{100}{143} = 0.7.$$

If on a bogie having the characteristic C*s* the strength $\delta s = 222$ kgr. per mm. (12 432 lb. per inch) we get

$$p = \frac{2\delta}{\delta_s} = \frac{200}{222} = 0.9,$$

because in this case *s* represents the two individual springs on one side of the bogie and it is necessary therefore to compare it with 2δ to bring the values C of the different methods of support to the same basis of comparison.

What do we learn from these curves? They show that :

The higher the value of the method of support characteristic C *given numerically in figure 19, the smaller is the factor* C *on which depends the change of wheel pressure* $\Delta P = C \delta X$.

For a given value of δ X, the method of support of the bogie can therefore exercise an appreciable influence on the static safety of running. This comparative value can be used as a scale of comparison of the static quality of the method of support.

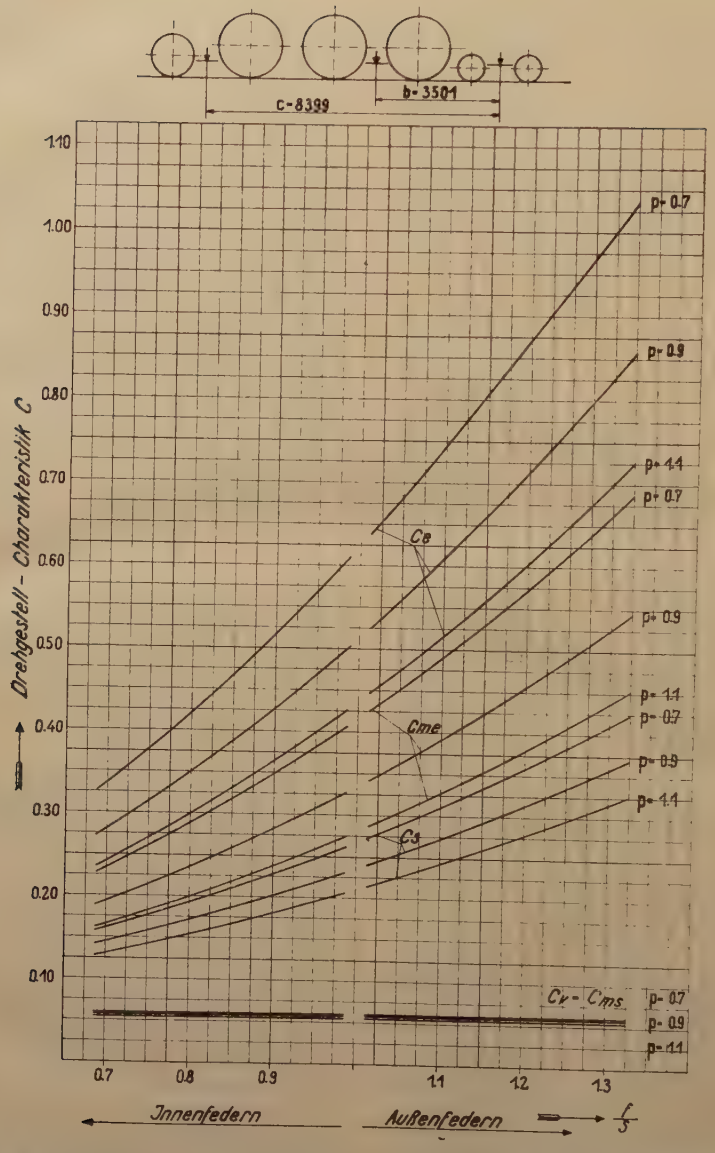


Fig. 19. — Value C 3 of the equation P = CδX for the bogie guiding wheel of a 4-6-2 locomotive.

Explanation of German terms:

Drehgestell-Charakteristik C = Bogie characteristic C. — Innerfedern = Inner springs. Aussenfedern = Outer springs.

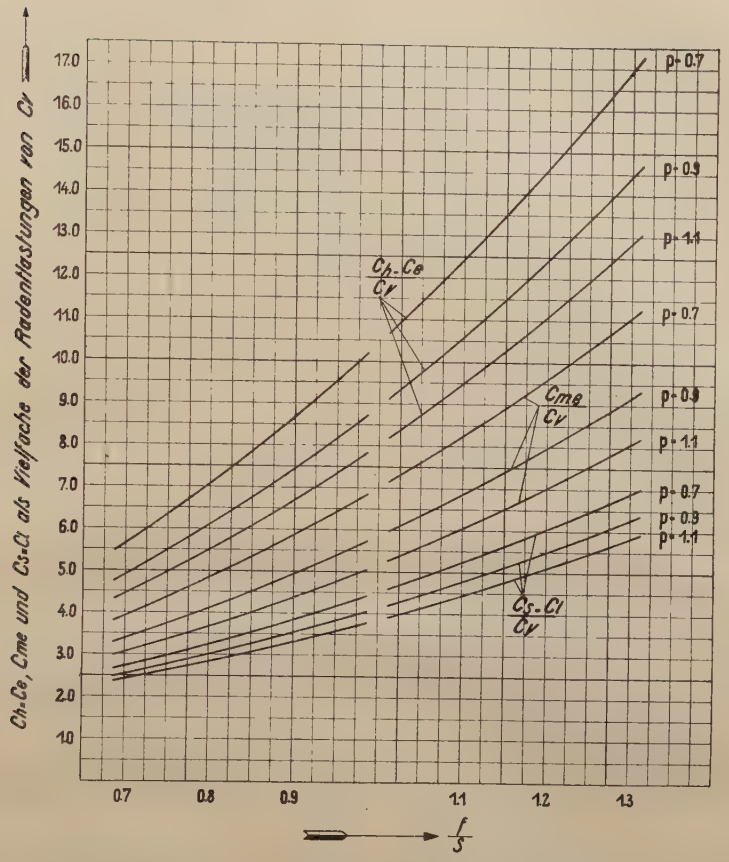


Fig. 20. — $Ch = Ce$, Cme and $Cs = Cl$, as multiples of the wheel load reductions of Cv.

Outside springs ($\frac{f}{s} > 1.00$) *cause greater changes in the pressure of the wheels than inside springs. The relatively soft bogie springs produce smaller changes of wheel pressures than stiffer springs.*

We are now in a position to arrive at close numerical values of the relations in question. We are also able to see clearly in what measure the support of the main frame on the bogie by a spherical centre is statically superior to the bilateral support; the values Ce drop to Cme, the values Cs to $Cms = Cv$. From the static point of view, the support Cv is the best method of all and in addition is independent of the position of the bearing springs whether between the wheels or outside them. If with the support in three planes on six points, Cv reaches a certain value, always small this is solely due to the front plane of support being lowered by $\frac{X}{4}$ mm. when the leading wheel of the left hand side drops X mm. and to the fact that the part of the sprung load, no longer sup-

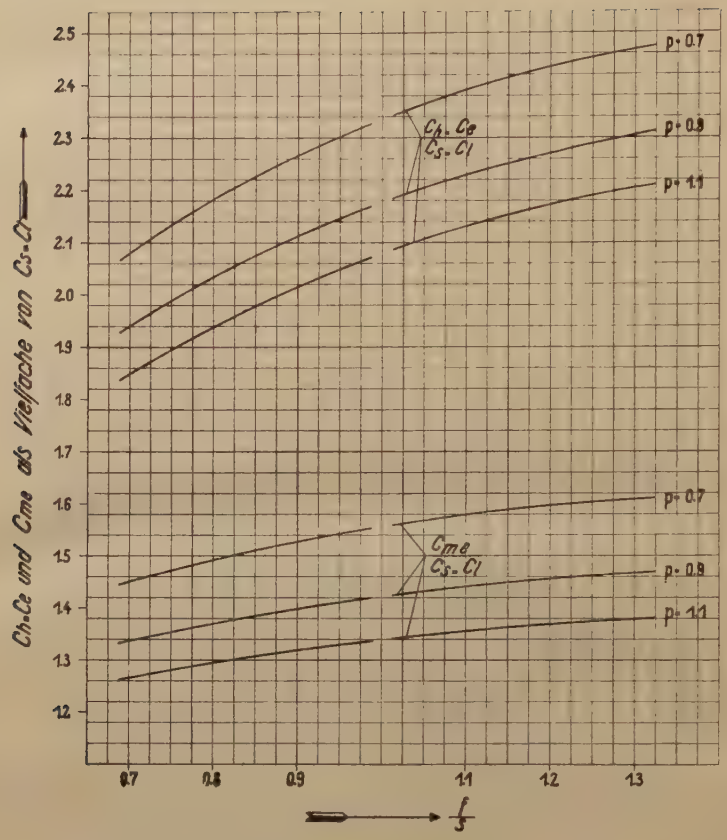


Fig. 21. — C*h* = C*e* and C*me* as mutiples of C*s* = C*l*.

ported after the lowering of the leading support A, is taken by the two other planes of support. The support is supposed to be in the middle transverse plane of the bogie. The equal reduction in load on all the bogie springs, expressed by $\frac{X}{4}$ mm. necessitates a corresponding reduction in load on the whole bogie. This is the great advantage of the support C*v*. Statically the bilateral support of the main frame on a bogie of this type is no worse in the position of the locomotive on which this theory is based, but in this position alone, than the statically ideal support by a single spherical centre at the centre of the bogie, seeing that the main frame with its wheels placed on the horizontal track exerts no transverse moment on the bogie frame. But when two, three, and finally all the pairs of wheels are on the irregular surface of the track, the central support of the frame on a spherical centre is statically better, even in the case of a bogie with a front equaliser, than the lateral support.

The diagrams of figures 20 and 21

are derived from those of figure 19; they represent different characteristics as multiples of C*v* and C*s*.

Compared to the support of the locomotive in three planes such as we have considered up to the present, the support in two planes on three or four points is much better as regards alterations in pressure $\Delta P = C\delta X$ of the directing wheel. For this reason, in this report, we have first of all dealt with the less favourable method of support in three transverse planes, used on 53.3 % of the steam locomotives mentioned in the replies.

The support on three points does not require any commentary or any recommendation. On the other hand, owing to the actual importance of the support on four points, it is also necessary to examine in the case of locomotives carried in two transverse planes, the characteristics of the bogies graphically, and to determine subsequently the numerical conditions by making use of the values of C of the two locomotives supported by the calculations, first of all in two, then in three transverse planes. In the notations, we add for example to the characteristic C*e* or C*s* the figure 2 or 3, according as the locomotive is supported in two or three transverse planes.

As now concerns this method of carrying a locomotive in two transverse planes only, the investigation by calculation has shown that the values C2 of the characteristics of the bogie are independent of the distance which separates the two transverse planes. They apply therefore to all locomotives with directing bogie supported in two transverse planes. Their numerical value only depends upon the total of the ratios between p and $\frac{f.}{s}$

Figure 22 shows in a graphical manner the values of C2. The calculation shows that these values are smaller than the values of C3 obtained for the same locomotive carried in three planes of support. The values C*v*2 and the values C*mv*2 and C*ms*2 become zero; with these methods of carrying the bogie and the locomotive there is no reduction in load on the leading wheel of the directing bogie. Consequently this method of support :

— bogie fitted with an equaliser at the leading end, with the main frame carried by the bogie on two lateral friction blocks (transverse knife edges), or

— bogie fitted with a transverse equaliser at the leading end and a spherical centre, or

— bogie fitted with longitudinal equaliser springs or a longitudinal equaliser at the leading end, and a spherical centre, with the main frame supported in two transverse planes only, is *statically* superior to all methods of support.

Figure 23 shows by how much the values C of the locomotive supported in three horizontal planes are greater than the values C of the same locomotive supported in two transverse planes. This figure shows that the method of support C*s* gives the best results, that the advantage is less with the method of support C*me* and least appreciable with the method C*e*, if instead of the support in three transverse planes, that in two transverse planes alone is selected. Within each of these groups of curves C, the advantage is greatest for a large p, i. e. with a relatively soft suspension of the bogie, and in addition for a smaller ratio $\frac{f.}{s}$ i. e. with inside springs of which the transverse distance apart is relatively small.

We will now explain the above statements by two examples following a method which is of general application. For this purpose we will select first of all the 4-6-2, series O 1 locomotive of the German State Railways supported in two planes on four points. In order to compare the values C that would be obtained if the same locomotive were supported on three points, we have calculated the values C3. The values that we have found are given on the left hand side of table 5 below. On the

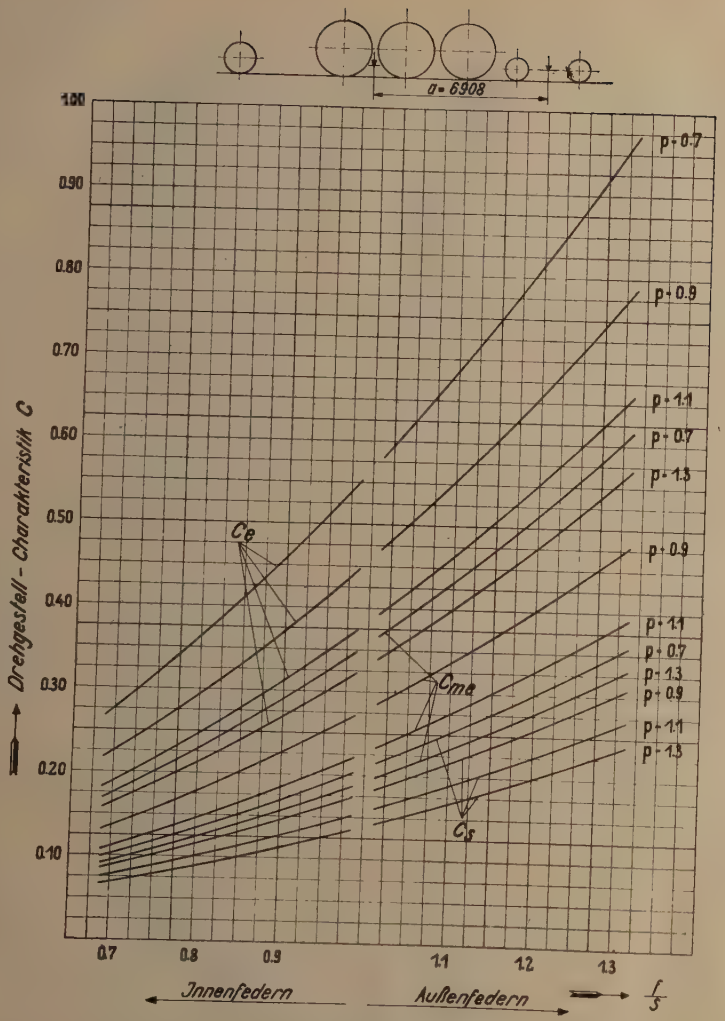


Fig. 22. — The values C 2 of the equation $\Delta P = C\delta X$

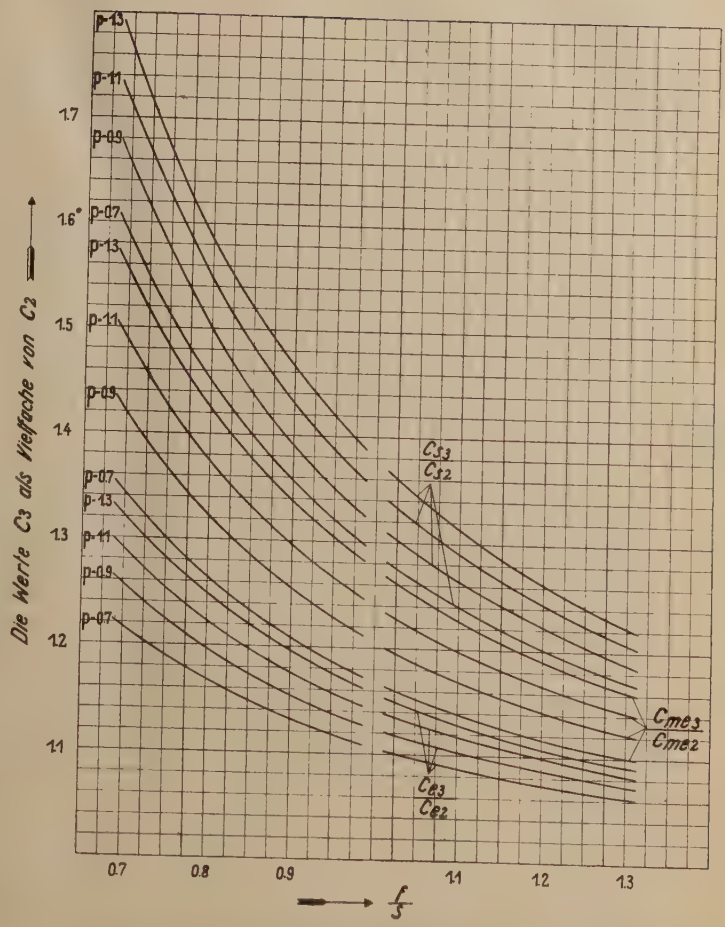


Fig. 23. — The values C 3 as multiples of C 2.

right side of this table opposite them the values of C2 and C3 have been given for the 4-6-2, series 18^5 locomotive of the German State Railways (S 3/6 of the former Bavarian State Railways), selected as the second example, which was supported in the first batches in three transverse planes, and which is supported in two transverse planes in the recently built engines.

TABLE 5.

Bogie characteristics and their multiples.	4-6-2 — Series 01 p = 1.335 when supported in		4-6-2 — Series 18^5 (S 3/6) p = 1.00 when supported in	
	2 transverse planes.	3 transverse planes.	2 transverse planes.	3 (old design) transverse planes.
Ce	—0.162	—0.213	—0.241	—0.297
Cs	—0.068	—0.120	—0.096	—0.154
$\frac{Ce2}{Ce3}$	0.76		0.84	
$\frac{Cs2}{Cs3}$	0.57		0.63	
$\frac{Ce3}{Ce2}$	1.32		1.23	
$\frac{Cs3}{Cs2}$	1.76		1.60	

We see clearly that the wheel of the directing bogie which finds itself in the depression of the track has less load removed from it with the method of support C2, sometimes much less, than in the case of the support C3. The proportion in which this static advantage is to be increased depends therefore only on the method of support of the bogie selected. On the 4-6-2, series 01 locomotive, the bogie of which has the support characteristic Cs, the support in two transverse planes results in a change of pressure on the wheels which is 43 % smaller than with the support in three transverse planes. Moreover the characteristic C2 of the 4-6-2, S 3/6 locomotive recently delivered, the bogie of which has the characteristic Ce, has reduced the amount of load taken off the wheels of the guiding bogies by 19 %.

It would be possible to obtain a still more favourable result from the static point of view if the locomotive supported in two transverse planes had been fitted with a bogie having the characteristic Cv, Cmv, or Cms, because we should then get $\Delta P = 0$, as we have already shown above in the general remarks. This is always in accordance with the theory of the position of the locomotive which is the point of departure of the whole of this investigation, i. e.: the leading pair of wheels of the bogie, but this pair of wheels

only, placed on the superelevation ramp leading from the curve to the straight.

As regards the calculations and practical investigations throwing light on the more or less satisfactory adaptability of certain types of bogies to superelevation ramps, and to irregularities in the track, the only information received was from the German State Railways. From this we have learnt that the calculated reduction in load on the guiding wheel of the standard bogie of the German locomotive having the support characteristic *Cs*, is about 3 to 3.5 % when leaving the minimum radius curve [R = 180 m. (9 chains), gradient 1 in 556] and that when passing over a depression in the track [a 20-mm. drop on a rail length of 2 m. (1 in 100 on a length of 6 ft. 6 3/4 in)], the reduction in the wheel load is as high as 6 %.

We have personally carried out, on the Karlsruhe Division of the German State Railways, many investigations into the changes of pressure of the directing wheels when running over rails on irregular surfaces. The investigations resulted in the above mentioned results among others. The influence of the gradients and the lengths of the superelevation on the determination of the maximum speed is examined in the part of our report dealing with the track. From the considerations just given obviously we should insist upon long transitions with a moderate inclination. For the minimum curve of 180 m. (9 chains) radius, the German State Railways has fixed the slope of the transition at 1 in 556 (1.8 mm. per m.). In practice steeper gradients are sometimes met. When relaying the track, or for other reasons, the superelevation is not always laid in throughout on the small inclination prescribed; it sometimes happens that short lengths are laid in with much steeper gradients. In the interest of safety it is therefore necessary that the work of levelling up the rails on double track lines should always be carried out against the direction of running, so that any such short length can be taken on a rising gradient.

There are hardly any connections on a 1 in 100 gradient within the length of a locomotive; steeper gradients are still possible on smaller lengths in the event of the road being lifted by frost. In order to show clearly and express numerically the differences of the effect of such and such a method of support of the reductions of load on the wheels and to be able to represent the processus of the changes of pressure on the wheels such as take place when the locomotive runs off the superelevation ramp, we have calculated the case of a 4-6-2, series 18^5 locomotive, a former Bavarian S 3/6 locomotive, supported in three planes and fitted with a *Ce* bogie, running down a superelevation ramp on a slope of 1 in 100. In order to carry out this investigation the locomotive was pushed in each experiment over a length equal to its wheel base on the slope and a calculation made of the amount of reduction of load on the directing bogie wheel going down the superelevation ramp when the wheels of the locomotive came exactly, one after the other, in line with the point of change of the gradient at which no rounding off was allowed. In order to ascertain the influence of the method of support of the bogie and the method of support of the whole of the locomotive in two or three transverse planes, the tests were repeated for the same locomotive supported in two transverse planes only. Finally the change of pressure of the leading wheel of the bogie was again calculated for the case in which the locomotive was fitted with a *Ce* bogie instead of a *Cs* bogie, and with the locomotive supported first of all in three and then in two planes. Table 6 below gives the results obtained, which are also represented in figure 24.

Everything we have dealt with in the general considerations and proved by nu-

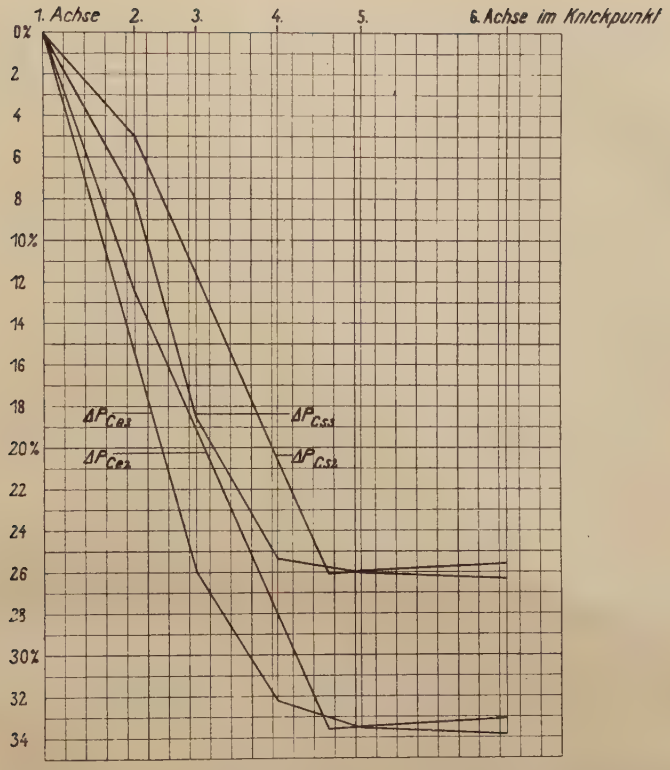


Fig. 24. — Variations of wheel pressure of the guiding pair of wheels of the bogie when leaving a superelevation incline of 1 in 100.

Note: Achse im Knickpunkt = Axle at point where gradient changes.

merical examples demonstrates how much better from the static point of view it is to support the sprung weight of the locomotive in two planes on three or four points than to support it in three planes. We are now in a position to interpret fully the meaning of the passages in italics in Chapter II; we can better understand the result of deciding in favour of one or the other method of supporting the main frame and its effects on the alterations of pressure of the leading pair of wheels of the locomotive. We see that 46.7 % of the steam locomotives mentioned in the replies are supported in two planes (30 % on three points and 16.7 % on four points), and 53.3 % in three planes, while on the electric locomotives the method of support in two planes, statically better, has been selected for a much greater number, that is 21.4 + 42.8 = 64.2 % of these locomotives.

We can also more clearly appreciate, from the diagrams of figures 19 to 23, the influence of the design of the bogies on the static safety of running, as we can very easily give numerical values to the distinctive characteristics. It is with this object that we have imagined the idea of the « bogie characteristic C ». It

is true that this notion is connected with a certain position, defined above, of the locomotive, this latter being placed on a straight and level line, only the left leading bogie wheel being in a X-mm. depression in the track. We explained at the same time that as regards the alteration of pressure of the leading guiding bogie wheel, this position produces exactly the same effect as when the second pair of wheels of the bogie leaves the superelevated curve to follow the first pair of bogie wheels which is already on the superelevation ramp and X-mm. lower. It is not permissible to generalise on such limited information, and we were therefore led to ask if this « bogie characteristic » really expressed the static quality of the design. This characteristic is, after all, only used to determine the change of pressure of the wheel for the sole position in which the locomotive, through a depression on one side of the track on the straight, has one of its two carrying wheels lower than the other, or has arrived at the end of the curve, with its leading bogie wheels on the down gradient of the superelevation ramp.

To elucidate the more general significance of the « bogie characteristic », among other reasons, we have given the results of the calculation reproduced in figure 24. We conclude therefrom as follows : when the locomotive runs onto the superelevation ramp, the load taken off the directing wheel at the leading end increases naturally with the difference in level, but the superiority of the method of support in two planes over that in three planes only lasts until the position of the change in gradient lies between the 4th and 5th pairs of wheels. Calculation shows that the change corresponds to the position in which the back plane of support is exactly above the point of change of gradient. But after a time the curves calculated for C2 become higher again, whatever the design of bogie, and lie above the curves calculated for C3 though but little, it is true. Even if continuing down the transition, the two methods of support (in two or three transverse planes) for a given type of bogie give practically equal reductions of wheel loads (cf. table 6), this does not alter the fact that, up to this position,

TABLE 6.

Method of supporting the locomotive.	When the					
	Ist	IInd	IIIrd	IVth	Vth	VIth
	pair of wheels is on a gradient change of 1 in 100, the load of the leading guiding bogie wheel is reduced by					
	%	%	%	%	%	%
Ce3	0	15.4	25.95	32.20	33.55	33.85
Ce2	0	12.5	19.13	28.10	33.50	33.10
Cs3	0	7.9	18.50	25.35	26.05	26.30
Cs2	0	5.0	11.63	20.60	26 00	25.60

the differences between the changes in pressure of the wheels, differences which depend essentially upon the value of C, existed for some time and sometimes reached very high values. The bogie characteristic, therefore, appears very suitable for use as an impartial basis of comparison when it is a question of appreciating the properties of the different types of bogies.

As far as the type of bogies themselves, considered in Chapter II, and which as

we have seen in this Chapter III, can statically influence very markedly the removal of the weight off the wheels which in turn depends chiefly on the method of supporting the main frame, we may sum up as follows :

A striking fact is that on the steam locomotives 35.7 % of the bogies have the *Ce* type of support, whereas only 10.7 % have the slightly better support *Cme*, and only 21.4 % the still better support *Cs*. The *statically* ideal support *Cv* is not represented at all. The position is different in the case of electric locomotives on which, in a proportionally greater number of cases, the statically superior method of supporting the main frame in only two transverse planes has been selected. Actually, among the electric locomotives with bogies with carrying pairs of wheels, 66.6 % have the *Cv* support and 16.7 % each of the *Cme* and *Ce* supports. Considered from the static point of view, the bogies with carrying pairs of wheels only, of the locomotives mentioned in the replies to the questionnaire, much better meet the conditions of maintaining as far as possible a constant load on the guiding wheels.

With the exception of the Java bogie of the Dutch East Indies State Railways, the bogies with a pair of carrying wheels and a pair of driving wheels are of the Krauss-Helmholtz type in which the driving pair of wheels is carried in the main frame. For the reasons given in Chapter IV this arrangement is considered to be the best.

The bogies of the tenders mentioned in the replies have the *Cm's* support in one case only; in all the other cases the *Ce* support is used. It should, however, be noted that only some of the administrations have given any information on this point.

In conclusion the following should be noted :

In order to have sufficient static load on the guiding wheels, the locomotives should be supported in two transverse planes and consequently on three or four points.

Not more than four points of support should be used on locomotives, unless their design makes it necessary.

If it is necessary to support them on five points, this should be done by the bogie being carried in a spherical centre bearing.

As immunity against derailment depends very largely on the bogie, it should be designed to take in account the method of support of the main frame in two or three planes and should provide the greatest possible guarantee against any difference in load on the four wheels, or at least on the leading pair, when running over irregularities in the track or over sections of the track in bad condition.

The question of knowing what designs should be used for bogies from the static point of view and what is the role of the ratio of the strength of the main frame bearing springs to that of the bogie springs can be easily setiled by means of the explanations and diagrams relating to the characteristics of the bogie.

The correct distribution of the load is not, however, sufficient by itself. Each individual pair of wheels must be able to adjust itself to hollows or high places in the track, and this result can only be obtained if each pair of wheels can describe a certain angle round about an axis parallel to the longitudinal axis, i. e. incline itself in the transverse plane of the vehicle. This condition is met if the flanges of the axle box guides, parallel at the middle only, are chamfered off at the top and the bottom, and if in addition there is a minimum play $x_{erf.} = h_1 \tan \alpha$ (cf. fig. 25), between the box and the guide. In this case the guiding allows the pairs of wheels to set themselves to an inclination equal to the value of the chamfer of the guides. If the play between the guide and the box is greater

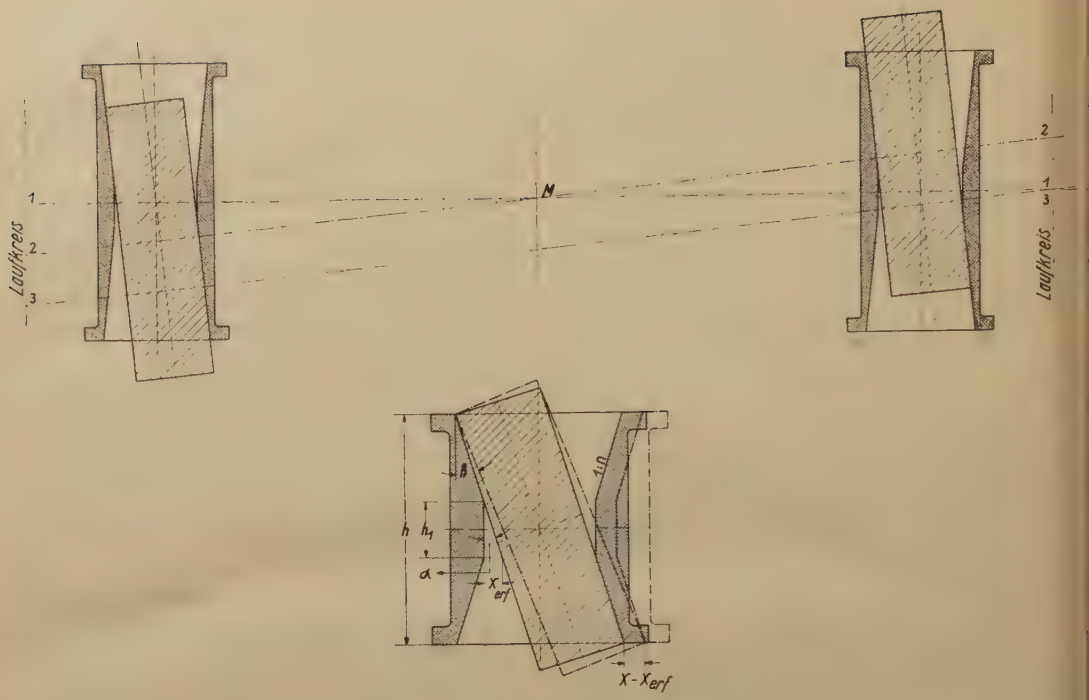


Fig. 25. — Possible inclination of axles.

than the calculated value of x_{erf} the axle can take an increased inclination which can be calculated from the formula :

$$\tan \beta = \frac{\frac{1}{n} \cdot h_1 + (x - x_{erf})}{h},$$

in which x is the real play and x_{erf} the minimum play required. The meaning of the other values is given in figure 25. With excessive play the guiding would become defective too quickly; in other words under the action of the continual blows and shocks, the play would soon increase to such an extent that it would have to be taken up.

The particulars received as regards the details of the steam and electric locomotives are not sufficient to enable us to know, except in rare cases, the arrangements made in this order of ideas on each locomotive. As far as can be ascertained from the information or drawings sent, the play between the axle box and guides, at right angles to the longitudinal centre line of the vehicle, varies between 1 and 2 mm. (5/128 and 5/64 inch), and the chamfers of the guides varies between 1 in 60 and 1 in 155. The German State Railways, in the case of their standard locomotives have 1 mm. (5/128 inch) play between the box and the guides; the flanges of the guides are inclined 1 in 60, starting 30 mm. (1 13/16 inches) above and below the centre line of the guide.

Table 7 shows, for 1 and 2 mm. (5/128 and 5/64 inch) play between the box and

the guides, the amount an axle can drop and how much it can be inclined horizontally, with the different chamfers of the flanges of the guides. The amount of drop on one side can be 12.4 to 25 mm. (15/32 inch to 63/64 inch), and the axle can take an inclination on the horizontal plane of 1 in 127.8 to 1 in 60.

In figure 25, the curves 1-1 correspond to the horizontal position, 2-2 to the rotation about M, and 3-3 to the drop on one side.

TABLE 7.

Possible drop of an axle, on one side.

Axle box guide inclined :	Possible drop on one side of axle, with, a 1 500 mm. (4 ft. 11 in.) distance between tread circles.	Possible inclination of axle on the horizontal plane.	Administrations and locomotive series to which these conditions apply.
1 : 60, x = 1 mm. (5/128 in.) play and h_1 = 60 mm. (2 21/64 in.)	25 mm. (63/64 in.)	1 : 60	German State standard locomotives.
1 : 60, x = 2 mm. (5/64 in.) play and h_1 = 60 mm. (2 21/64 in.)	30 mm. (1 3/16 in.)	1 : 49.75	German State standard locomotives with (x = 2 mm.) play in axle boxe guide.
1 : 90, x = 1 mm. (5/128 in.) play and h_1 = 60 mm. (2 21/64 in.)	17.6 mm. (1 3/8 in.)	1 : 84.7	German State, also former Baden (State) 4-6-2 locomotive.
1 : 90, x = 2 mm. (5/64 in.) play and h_1 = 60 mm. (2 21/64 in.)	19.9 mm. (25/32 in.)	1 : 67.5	German State, also Baden. 4-6-2 locomotive with (x = 2 mm.) play in axle box guide.
1 : 120, x = 2 mm. (5/64 in.) play and h_1 = 80 mm. (3 5/32 in.)	20.6 mm. (1) (53/64 in.)	1 : 76.7	Finnish State. Carrying wheels of 4-6-0 locomotive, series H 8 and H 9.
1 : 120 x = 1 mm. (5/64 in.) play and h_1 = 80 mm. (3 5/32 in.)	15.6 mm. (1) (3 9 64 in.)	1 : 101.3	Finnish State. Carrying wheels of 4-6-0 locomotive, series H 8 and H 9, when x = 1 mm. was used.
1 : 155, x = 2 mm. (5/64 in.) play and h_1 = 80 mm. (3 5/32 in.)	16.8 mm. (1) (43/64 in.)	1 : 94	Finnish State. Driving and coupled wheels of 4-6-0, series 8 locomotive.
1 : 155, x = 1 mm. (5/128 in.) play and h_1 = 80 mm. (3 5/32 in.)	12.4 mm. (1) (1/2 in.)	1 : 227.8	Finnish State. Driving and coupled wheels of 4-6-0, series 8 locomotive, when x = 1 mm. was used.

(1) With a distance of 1.58 m. (5 ft. 2 13/64 in.) between tread circles, to conform with the 5-foot gauge of the Finnish State Railways.

To sum up the following may be said : *The possible inclination of the axles in the axle box guides must be at least sufficiently large for the axles not to stress the frame by torsional stresses when the vehicle runs over sections of*

the track with irregular surfaces. Owing to the unavoidable differences in level of the track, the axles ought to be able to take up an inclination of at least 1 in 60 to the horizontal.

We have shown earlier on the importance that must be attached from the point of view of maintaining the static safety of running to a proper relationship between the strength of the springs between the leading guiding pair of wheels, and those in the main frame of the locomotive. The springs must also be able to damp out and destroy the vertical shocks as quickly as possible by internal friction.

As the acceleration of shock becomes very great at high speed and in addition the work of the shock is proportional to the square of the acceleration of the shock (*Bulletin of the Railway Congress*, 1932, p. 234), a preliminary condition of good running of a vehicle at high speed is good suspension gear.

Laminated springs are used almost exclusively to carry locomotives. This is well justified. They have the great advantage over other types of springs of quickly destroying the work of shock by the friction between the different plates of the spring, and consequently absorb very rapidly the vibrations of the locomotive due to outside causes. Recent investigations by the German State Railways have shown that laminated springs only take up very well damped oscillations, are almost aperiodic, and have no tendency to vibrate through resonance.

The softness or the hardness of a spring depends upon the « strength » of the spring; we have designated it above by S : this is the load required to deflect the spring one millimetre. A locomotive spring on which $\delta = 100$ can be considered as a « soft » spring.

A soft spring gives steady and pleasant running. It is known however that limits exist with regard to the selection by the design and by the position of the buffers. The strength of the springs can be calculated by means of the formula :

$$\delta = \frac{1}{3} \cdot \frac{nbh^3}{l^3} \cdot E,$$

in which n, b and h represent the number, the width and the thickness of the leaves, δ the distance of the load from the centre line of the spring, and E the modulus of elasticity. It is true that the friction between the plates has some considerable influence upon the strength of the springs. If this friction is too great the spring gear becomes harder and the springs do not act except under considerable variations of load. The friction can be maintained within certain limits by using thick plates, and consequently by reducing their number, also by lubricating the surfaces in contact.

Let us see what has been done as regards the question of the *spring gear of the 30 steam locomotives considered and the electric locomotives.*

The spring gear of the steam locomotive bogies considered is almost always harder than that of the driving and coupled pairs of wheels. We have shown why this is almost always the case. As we have seen when considering the « support characteristics » of the bogie, the ratio of p should be as large as possible; consequently the bogie springs should be as soft as possible. The tests that we have carried out with soft laminated springs in a single spring gear have given unsatisfactory results, in the sense that these springs often broke on the sections of line in not quite the best order. Considered as a whole, the strength of the individual springs on the bogies of the steam locomotives mentioned in the replies lie between 108 and 249 kgr. per mm. (6 048 and 13 944 lb. per inch), and those of the springs used with the drop equalisers between 125 and 270 kgr. per mm. (7 000 and 15 120 lb. per inch).

For example the proportional strength of the bogie springs with drop equalisers

Cs of the standard German locomotives, series O1, is

$$p = \frac{2\delta}{\delta_s} = \frac{2 \times 167}{250} = 1.337,$$

that of the *Cm's* bogies of the 4-8-2 locomotive of the Madrid-Saragossa-Alicante Railways

$$p = \frac{2 \times 133}{270} = 0.985$$

(lowest value of the locomotives mentioned the replies); and those of the *Ce* bogie of the 4-6-2, series 18[5] locomotive of the former Bavarian State Railways

$$p = \frac{\delta}{\delta_e} = \frac{150}{149} = \text{about } 1.0.$$

The usual arrangement used under passenger carriages consisting of a double suspension formed by the laminated springs each with its two helical springs is also found on some of the steam locomotive bogies mentioned in the replies; such is the case in the *Ce* bogies of the 4-6-2, series 24 locomotives of the Dutch East Indies State Railways, which have a $\delta_e = 70.9$ kgr. per mm. (3 970 lb. per inch) and a

$$p = \frac{118}{70.9} = 1.665$$

(maximum value), as also for the *Cm'l* bogies of the 4-6-0, series 3900 locomotives and the bogies of the 4-6-4, series 6100 locomotives of the Netherlands Railways, in which

$$p = \frac{143}{97.2} = 1.470$$

for the 3900 series and

$$p = \frac{125}{97.2} = 1.287$$

for the 6100 series. The extraordinarily good result of high values of p is brought out by our detailed explanations and by the diagrams.

The importance of soft springs as affecting the vehicle and the passengers or the staff has long been shown by experience. It is a question which has frequently been dealt with in technical literature, in, among other works, *Eisenbahntechnik der Gegenwart, Band Lokomotiven,* 1912). By using Uebelacker's method we have expressed it as a function of the change of pressure of the guiding wheels and believe we have thereby lifted it above any approximate or empirical appreciation.

To sum up, the following may be said on the subject of the suspension of bogies :

The proportional strength p of the *Ce* bogie springs with single springs above each axle box lies between 0.57 and 1.00. In one case only of the *Ce* type of support (in locomotive 4-6-2, series 24, of the Dutch East Indies State Railways) the use of a double suspension made it possible to get for p the value of 1.665.

The proportional strength p of the bogie springs *Cs* with drop equalisers lies between 0.985 and 1.665, and that of the *Cl* bogies with longitudinal equalisers and double suspension, between 1.287 and 1.470.

In the case of the driving and coupled pairs of wheels, springs with a strength of 92.6 to 167 kgr. per mm. (5 185 and 9 356 lb. per inch) are used; the only exception is the Spanish 2-8-2, series 4 500 locomotive, with $\delta = 285$ kgr. per mm. (15 960 lb. per inch).

The spring gear of the trailing pair of carrying wheels is usually of the same hardness as that of the coupled wheels; in some cases it is slightly softer, for example on the 4-6-2 locomotive of the Madrid-Saragossa-Alicante Railways, or a little harder, as for example on the 4-6-2, series 24 locomotive of the Dutch East Indies State Railways and on the 4-6-2, series 3 000 locomotive of the North of Spain Railways.

As regards the *springing of electric locomotives,* we have only received few particulars. As a rule the springs of

the carrying and driving wheels have the same strength; in many cases, however, the spring gear of the leading carrying wheels is particularly soft by the addition of helical or volute springs. This double suspension is used, for example, on the 2C2, series 06 locomotives of the German State Railways for the bogie, and on the 1 Do 1, series E 16 and E 17 of the German State Railways and the Java 1 Do 1 locomotive for the leading pair of carrying wheels of the bogie.

The strength of the tender springs is generally very high [as much as 400 kgr. per mm. (24 400 lb. per inch)], in order that the difference of buffer height between the tender under load and the tender empty does not become too great. As far as can be judged from the information supplied, in the case of the tenders considered, soft springs [δ = 80 to 100 kgr. per mm. (4 480 to 5 600 lb. per inch)] are only found on the German State Railway tenders; as we have already mentioned in Chapter II some of these tenders have been fitted with double suspension.

The dimensions of the laminated bearing springs are given in table 8 below :

TABLE 8.

—	Length of blades of bearing springs		Width of blades of bearing springs		Thickness of blades of bearing springs		Numb of bla
	from	to	from	to	from	to	from
	mm. (*ft.-in.*)		mm. (*inches*)		mm. (*inches*)		
Bogie (leading carrying wheels) . . .	650 (2 ft. 1 5/8 in.)	1227 (4 ft. 5/16 in.)	75 (2 61/64 in.)	127 (5 in.)	8 (5/16 in.)	16 (5/8 in.)	6
Driving and coupled wheels	907 (2 ft. 11 45/64 in.)	1400 (4 ft. 7 1/8 in.)	90 (3 35/64 in.)	120 (4 23/32 in.)	10 (25/64 in.)	16 (5/8 in.)	9
Trailing carrying wheels	700 (2 ft. 3 35/64 in.)	1459 (4 ft. 9 7/16 in.)	90 (3 35/64 in.)	120 (4 23/32 in.)	8 (5/16 in.)	16 (5/8 in.)	9
Tender wheels . . .	650 (2 ft. 1 5/8 in.)	1200 (3 ft. 11 15/64 in.)	80 (3 5/32 in.)	127 (5 in.)	10 (2 5/64 in.)	16 (5/8 in.)	6

The German State Railways prefer long wide springs with 13 to 16 mm. (33/64 to 5/8 inch) thick blades; the number of blades is reduced as much as possible in order that the friction is not excessive. Other administrations prefer thin blades and a larger number.

With only one exception (the 2-8-2, series 4500 locomotive of the North of Spain Railways), *helical and volute springs* are not used on the locomotives mentioned in the replies except for use *in conjunction with laminated bearing springs.*

Helical and volute springs can be designed to give any desired softness; they absorb the shock very well, but are liable to set up vibrations which can last a considerable time, as with such springs the energy of vibration is not absorbed by outside work as in the case of laminated springs. This is the disadvantage of such springs. They damp out shocks very well, and a vehicle

fitted with them runs very smoothly, but they do not stop vibrations as quickly as laminated springs, which is objectionable for coaches. *As supplementary suspension for laminated springs; helical and volute springs are often used,* as we have already mentioned, on electric locomotives, steam locomotives and tenders. They appear moreover to have justified their use in this connection. In every case in which numerical values have been given, the strength of the laminated springs is greater than that of the helical springs.

The *loads on the locomotive bearing springs* are lower than those usually given to the bearing springs of passenger carriages. The maximum static load given in the replies varies between 45 and 58 kgr. per mm² (28.6 and 36.8 Engl. tons per sq. inch). In the case of spring steel the German State Railways lay down a tensile strength of 85 kgr. per mm² (54 Engl. tons per sq. inch), with 12 % elongation (on a 200-mm. [7 7/8 inches] length), and the other administrations a tensile strength of 75 to 85 kgr. per mm² (47.6 Engl. tons per sq. inch), with 12 to 14 % elongation. The hardened steel should have a strength of 140 kgr. per mm² (88.9 Engl. tons per sq. inch) (German State Railways, Dutch East Indies State Railways and Norwegian State Railways), 5 % elongation on a 200-mm. (7 7/8 inches) length, and a yield point (Streckgrenze) of 110 kgr. per mm² (69.8 Engl. tons per sq. inch).

Information relative to particular results obtained with the springs or to tests carried out in this field have not been sent by any administration.

CHAPTER IV.

Guiding the vehicles and devices intended to make it easier for the vehicles to run through curves and turnouts.

In the preamble to the preceding chapter, we have seen that a wheel permanently guaranteed against the load on it being reduced can in no case be pushed clear of the track by the guiding pressure Y acting transversely to the rail, from the inside to the outside, provided of course it carries sufficient load on it. By using Dr. Uebelacker's method it is possible without much difficulty to ascertain the changes of pressure of the wheels which occurs when running over irregularities of the track (superelevation inclines), especially those of the leading guiding wheel, the most important wheel of the whole of the wheels. We ourselves have added to Dr. Uebelacker's theory the relations which exist between the methods of support of the main frame in two or three transverse planes and the pressures on the guiding wheels of the bogie, according to the type of the latter, taking into account the various types of suspension of the main frame and of the bogie. We are then able to ascertain beforehand the static effects which will result from the decision adopted as regards the method of support and the lay-out of the spring gear. But surface irregularities of the track are found not only in transitions before and after curves but also on the straight at places where there is a depression or a high place on one side; here again these sections must be passed over, especially by high-speed locomotives, without any effect upon the proper guiding and safety of running of the locomotive. As we have only just begun to realise through calculations and tests the true dynamic effects of sections of line of this kind, it is desirable to use the best possible spring gear and method of support. Fortunately all the necessary knowledge as far as purely static relations are concerned is now available.

The question is less simple as regards the static guiding pressure Y. This is the pressure between the leading guiding wheels running on the outer rail and this rail transversely from the inside face to the outside. Y constitutes with

the friction at this point the ultimate turning effort P. This force directs the vehicle and causes it to turn about the point which, with Dr. Uebelacker (*Organ*, 1903), we have designated as the « friction centre ». The position of the friction centre determines the position of the vehicle in the track. The turning effort P of the rail and the guiding pressure Y are connected together (cf. Dr. Heumann « *Organ* », 1913 and 1928, « Zum Verhalten von Eisenbahnfahrzeugen in Gleisbogen » (1), by the formula $P = Y + \mu Q \cos \varepsilon$, in which P and Y have the above mentioned meaning, μ is the coefficient of sliding friction of the wheels, Q the vertical pressure of the wheel on the rail, and ξ the angle which the direction μ Q makes with Y (cf. also *Verkehrstechnische Woche*, 1929). Y determines the danger of derailment of the wheel in contact. Its value is given by the equations of equilibrium between the external forces. It is usual to express the value of Y as a multiple of the average pressure Q of the wheel. The greater the lever arm with which Y is applied, the smaller under equal conditions is the value of Y. We have shown in Chapter I that this lever arm is not identical with the guided length but care must be taken to make the guided length as long as possible, as the said lever arm is proportional to it. Constructional considerations limit the liberty of choice. Other difficulties are due to changes in the coefficient of friction μ' which determines the value of the transverse sliding friction of the wheels in movement. In the *Organ für die Fortschritte Des Eisenbahnwesens*, No. 19 of 1930, there is an article on the value of μ', determined by tests, for wheels other than driving and coupled wheels. For greater safety when calculating the value of Y, the most unfavourable values that can occur under the expected striking angle and the conditions of friction should be used. Fortunately the tests carried out to date on models, in order to determine the values μ' of the driving wheels, make it possible to see that these values are smaller than the values μ' of the carrying wheels. The reflection that part of the friction due to adhesion is already absorbed by the peripheral force supports this result.

The principal difficulty consisted from the begining in knowing exactly the suitable load to prevent the guiding wheel from climbing up the side of the rail, i. e. determining numerically in units of Q (average weight of the wheel) the value of Y which, if exceeded, would result in unstable equilibrium, and consequently involve risk of derailment. The lengthy heading of this statement could have taken the very concise form Y : Q = ? if this method of expression was generally known and sufficiently intelligible, as it is this which is in question. All we have contributed to this study in the ultimate analysis has only been done with the object of clearing up the question of the steps necessary to keep the ratio Y : Q below a certain limiting value, so as to safeguard the safety at high speeds.

The value of Y : Q is generally considered as the criterion of the safety against derailment. Some, and they are many, consider that the distance of the turning effort to the friction centre, called by Jahn the polar length (Jahn : « Der Lauf von Eisenbahnfahrzeugen durch Gleiskrummungen » (1), Berlin 1927) should also be taken into consideration. This opinion appears justified as it amounts to taking into account the distribution of the masses in the vehicle, and consequently the action of the masses which we will briefly consider at the end of this chapter. If we

(1) « The behaviour of railway vehicles on curves. »

(1) « The passage of railway vehicles through curves. »

understand that in this Y is included the guiding pressure of the masses nothing opposes to this criterion being maintained, and this we have done.

For many years, railway engineers of all countries have been occupied with this question. To the best of our knowledge they have all developed the fundamental equation :

$$\frac{Y}{Q} = \tan(\alpha - \rho) = \frac{\tan\alpha - \mu}{1 + \mu\tan\alpha}$$

in which Y is the static guiding pressure, α the angle of inclination of the side of the wheel flange, ρ the friction angle ($\tan\rho = \mu$) and μ the coefficient of friction. For an inclination of $\alpha = 60°11'\ 51''$, we get $Y : Q = 5.7\mu$. We will suppose for the moment that Y has a given value, i. e. the value calculated for the conditions of equilibrium of the external forces, applied to the vehicle and its wheels, during *continuous* movement through a curve of given conditions. We will show later on what must be done to reduce Y to a minimum.

In the equation of equilibrium $\frac{Y}{Q} = \tan(\alpha - \rho)$, *mentioned above, the striking angle , i. e. the angle which the guiding wheel makes with the guiding rail, is not mentioned.* There has always been this missing quantity in this formula. Like Marié in his treatise « Theory of derailments, profile of tyres, » everyone was satisfied with establishing numerical tables of Y : Q for different α and μ and to deduce therefrom conclusions concerning the angles of slope of the flange. If other considerations were not taken into account the angle of inclination $\alpha = 90°$ would be that which would best assure immunity against the guiding wheel climbing. Nobody uses it, because it would result in excessive wear of the wheels and of the sides of the rails. Investigations made by the German Railway Association which had as their object the profile of the tyres from the double point of view of freedom from derailment and wear, have succeeded, in taking into account the striking angle in the equation of equilibrium. When a wheel making contact with the rail is in a position to climb the rail there is in this case contact at a single point, although two-point contact between the tyre and the rail may have occurred a little before. According to Dr. Wagner, the safety against derailment will still exist if

$$\frac{Y}{Q} \leqslant \frac{\sqrt{1 + \frac{h^2}{l^2}}}{\mu + \operatorname{cotan}\alpha}$$

In this equation :

h is the height of the point of pressure of the flange below the top of the rail,

l is the overlap of the flange i. e. the length of the projection on the longitudinal direction of the rail of the distance from the point of pressure of the flange to the point of support of the wheel (h and l depend therefore on the value of the striking angle τ),

μ is the coefficient of friction, and

α the angle of side inclination of the flange.

With the angle of side inclination $\alpha = 60°\ 11'\ 51''$ used by the German Railway Association, the point of pressure of the flange is situated at 9 mm. (23/64 inch) in round numbers below the rail for all diameters of wheels in use, from 850 mm. (2 ft. 9 1/2 in.) up to 2.100 m. (6 ft. 10 43/64 in.), and between striking angles of $\tau \leqq 0°$ to $2°$ which are found in practice. This is the result obtained from the graphical investigations and calculations we have made. According to these investigations, the value of the overlap l of the flange can be calculated for new wheels by means of the formula :

$$l = (r + h)\tan\alpha\tan\tau$$

in which

r is the radius of the tread circle of the wheel;

α the striking angle, and

h and α have the meanings given above [cf. also Boedecker, « Die Wirkungen zwischen Rad und Schiene (1), page 36, formula 22].

When $\alpha = 60° 11' 51''$, we get $h = 9.3$ mm. (49/128 inch) in round numbers 9 mm. (3/8 inch). Every other angle of inclination gives another value of h. The values of l calculated by means of this formula agree very closely with those obtained by the much more laborious method based on calculation alone, which Professor Jahn has given in the *Zeitung des Vereins Deutscher Eisenbahnverwaltungen*, 1917, page 427. For example, if $\tau = 1°$, $\alpha = 60° 11' 51''$, $r = 1$ m. (3 ft. 3 3/8 in.) and $\mu = 0.16$, we get : $l = (1\,000 + 9) \tan 60° 11' 51'' \tan 1° = 30.7$ mm. (1 7/32 inches) and

$$\frac{Y}{Q} = \frac{\sqrt{1 + \left(\frac{9.0}{30.7}\right)^2}}{0.16 + \operatorname{cotan} 60° 11' 51''} = 1.422.$$

Table 9 below and the graphical representation (fig. 26) give the relations in question for wheels of 2 m., 1.5 m. and 1 m. (6 ft. 6 3/4 in., 4 ft. 11 in., and 3 ft. 3 3/8 in.) diameter of the tread with an angle of inclination $\alpha = 60° 11' 51''$. We have made the same investigations for other angles of inclination.

TABLE 9.

Diameter of tread circle. mm. (*ft.-in.*)	Striking angle. τ	Overlap of wheel flange in mm. (*in inches*).	Coefficient of friction μ	Y : Q
2 000 (6 ft. 6 3/4 in.)	0° 15'	7.7 (0.303)	0.15	2.13
—	0° 30'	15.4 (0.606)	0.195	1.51
—	1°	30.7 (1.208)	0.21	1.33
—	1° 30'	46.1 (1.814)	0.21	1.30
—	2°	61.5 (2.421)	0.21	1.29
1 500 (4 ft. 11 in.)	0° 15'	5.8 (0.228)	0.15	2.56
—	0° 30'	11.6 (0.456)	0.195	1.65
—	1°	23.1 (0.909)	0.21	1.37
—	1° 30'	34.7 (1.366)	0.21	1.32
—	2°	46.3 (1.822)	0.21	1.30
1 000 (3 ft. 3 3/8 in.)	0° 15'	3.9 (0.153)	0.15	3.50
—	0° 30	7.8 (0.306)	0.195	1.99
—	2°	15.5 (0.610)	0.21	1.48
—	1° 30'	23.3 (0.917)	0.21	1.37
—	2°	31.0 (1.220)	0.21	1.33

We see that safety against derailment is very great when the striking angle is small. As the striking angle increases this degree of safety falls off rapidly at first, but then decreases very little in the region of quite large striking angles and approaches asymptotely the value $\frac{1}{\mu + \operatorname{cotan} \alpha}$

(1) The interaction between wheels and rails.

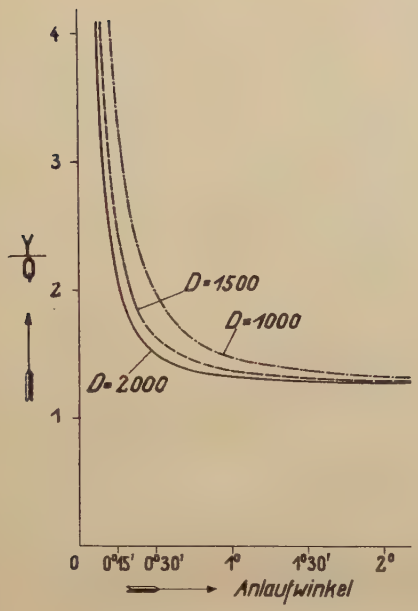


Fig. 26. — Safety coefficient $\frac{Y}{Q}$ as a function of the striking angle.

Note: Anlaufwinkel = Striking angle.

For μ we have selected the average of the values of Labrijn indicated in the *Organ*, 1931, page 391. The table of the values Y : Q and the diagrams would give other values with $\mu = 0.25$ for $\tau = 2°$; we would get for example :

Diameter of the wheel in mm. (in ft. and in.)	Y : Q
2 000 (6 ft. 6 3/4 in.)	1.228
1 500 (4 ft. 11 in.)	1.238
1 000 (3 ft. 3 3/8 in.)	1.266

The limiting value of the ratio Y : Q depends upon the diameter of the tread circle, the striking angle of the guiding wheel, the coefficient of *friction* μ *and the angle of inclination of the side face of the flange.* A large diameter of the wheel, a large striking angle, large coefficients of friction diminish the safety against derailment; on the other hand, a large angle of inclination of the flange incrases immunity against derailment.

Flange angles exceeding about 60° are nevertheless used by few railway administrations. This is due first of all, as we have already pointed out, to the fact that tyres with greater flange angles result in the rail and tyre wearing more quickly and also set up greater resistances when running through curves. According to Dr. Ing. Bäseler, the resistance due to curves increases by 140 % when a flange angle $\alpha = 75°$ instead of $\alpha = 60°$ is used (*Zeitung des Vereins Deutscher Eisenbahnverwaltungen*, 1929, page 372 : « Das Geheimniss der freien Lenkachse ») (1). The flange angle has therefore importance economically, so that it is necessary to find out how to conciliate in a satisfactory manner the two diametrically opposed conditions governing the angle α, i. e. sufficient safety against derailment and reasonable wear.

The tyre profiles designed or used by the administrations covered by the present report are reproduced in figure 27. We have calculated the inclination of the side face of the flange for the different profiles and have given the results in table 10.

Investigations made by the Hungarian Railways under the Divisional Engineer, Mr. Rónai, and on the German State Railways by Dr.-Ing. R. P. Wagner, in connection with this question, for the German Railway Association, have shown that *with a suitably designed profile of the tyre, and especially of the radius between the tread and the flange, the angle of the flange can, contrarily to what has been deduced previously from the equation* $\frac{Y}{Q} = \tan(\alpha - \rho)$, *be made less than 60° and consequently wear can be reduced without,* as Mr. Rónai and Dr. Wagner have shown, *the safety*

(1) The secret of the free radial axle.

TABLE 10.

On the	the	has an angle of inclination of
Administrations members of the German Railway Association	Full wheel flange.	60° 11' 51''
German State Railway Co. . . .	Wheel flange with 5 mm. (25/128 inch) ground off.	60° 42' 53''
German State Railway Co. . . .	Wheel flange with 10 mm. (25/64 inch) ground off.	59° 18' 0''
German State Railway Co. . . .	Wheel flange with 15 mm. (19/32 inch) ground off.	60° 29' 23''
Dutch Indies State Railways . . .	Full wheel flange for locomotives and coaches.	60° 4' 17''
Finnish State Railways	Full wheel flange.	64° 2' 13''
Madrid-Saragossa-Alicante Railways .	Full wheel flange.	72° 16' 32''
Do.	Thinned down wheel flange.	73° 44' 24''
North of Spain Railways	Full wheel flange for locomotives.	65° 6' 0''

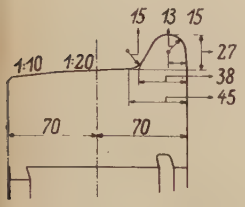
against derailment becoming less than when $\alpha = 60°$. The reason for this is that if the lateral force Y corresponding to the flange angle becomes greater, on the other hand the overlapping portion of the flange becomes smaller. From the purely static point of view, *when using a tyre profile designed on the principle of single point* contact and with a flange angle of 54°, the investigators consider that P : Q = 1 would still give sufficient security against derailment so much the more as, for the usual maximum speeds and the usual height of the centre of gravity of the locomotives, as with the usual superelevations of the outer rails on curves, it can be shown that to striking pressures Y which would mean the turning over of the vehicle, correspond the values : Y : Q much smaller than 1. Tests on wheels of this kind are in hand.

Consequently the builder should design the details of the running gear of the vehicle in such a way that the guiding pressure and the pressure on the guiding wheel should between themselves have a ratio such that the downwards slip of the guiding wheel on the side face of the flange should be made certain. With the flange angle of $\alpha = 60°$ approximately, this condition is still assured, as we mentioned above, if $Y : Q = 5.7\,\mu$; it is true that the striking angle has not been taken into account. This case occurs with a flange angle of 54° if Y : Q = 1.

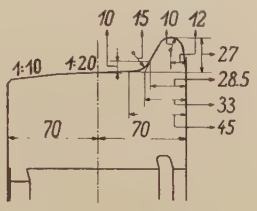
For more than thirty years it has been known that high-speed locomotives are more definitely guided by bogies with two pairs of carrying wheels, with the centres given side play, placed between the pairs of wheels, and with transverse spring gear. The reasons therefore have been explained on many occasions (cf. for example, *Eisenbahntechnik der Gegenwart,* Vol. I : Steam locomotives, 1912. First part : « Running on straight alignments and on curves »). We know that this is due to the relatively small mass of

German Railway Association.

Full wheel flange.

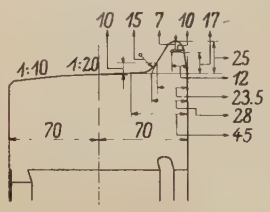


Flange thinned down 5 mm. (25/128 inch).

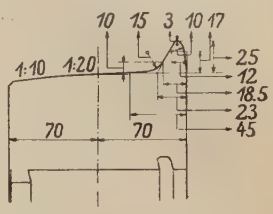


German State Railways.

Flange thinned down 10 mm. (25/64 inch).

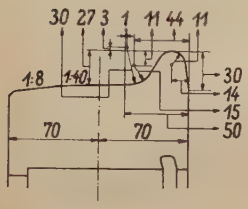


Flange thinned down 15 mm. (19/32 inch).

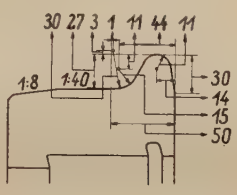


Dutch Indies State Railways.

Full wheel flange for locomotives.

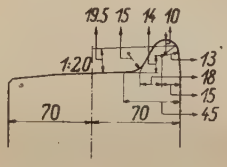


Full wheel flange for carriages.

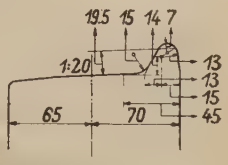


Netherlands Railways.

Profile B.

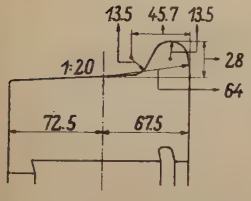


Profile C.

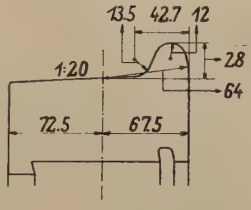


Finnish State Railways.

Full flange.

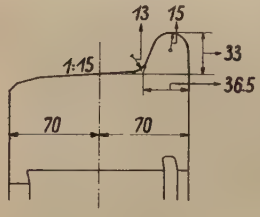


Thinned down flange.

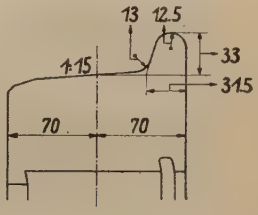


Madrid-Saragossa-Alicante Railways.

Full flange.



Thinned down flange.



North of Spain Railways.

Full wheel flange for locomotives.

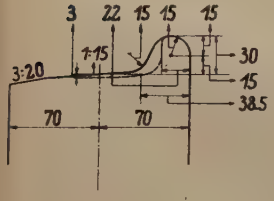


Fig. 27. — Tyre profiles.

the bogie and its ability to absorb elastically the inevitable transverse shocks set up by irregular changes of direction on straight alignments and on curves. Dr. Uebelacker (cf. *Organ*, 1930, p. 272) points out that *the principal object of the centring springs is not to restore the bogie to the central position but to give elastic guiding.* When the guiding wheel comes into contact with the side face of the rail at an angle, the wheel tends to change its direction. Nothing is more suitable for building up the force required to generate the work contained in the deflected vehicle than the centring springs. The limited mass of the bogie facilitates this processus taking place readily. The main mass of the locomotive, if we may be permitted to make this comparison, only receives a distant echo, more or less strong, of all the transverse shocks which may upset good running. The oscillograms relative to transverse movements of these carrying pairs of wheels show the extraordinary facility with which the more or less serious sinuous movements are set up by changes of direction or of level of the rails. From the most recent tests carried out by the German State Railways, changes of gauge of one or two millimetres are sufficient to set up these sinuous movements, even on curves. The graphs relative to the striking pressures generally have the form of a saw, at least this is so with the curves that we had in front of us when drawing up this report. This fact is certainly due to a large extent to the elastic flexibility of the rail and of the vehicle in the transverse direction, but also to a considerable extent, to the very feeble resistances opposing the transverse movements caused by small changes of direction which we have calculated to be quite a small number of minutes of arc. As regards the coefficients of friction of the transverse sliding movement of the running wheels, we have estimated these as equal in this region of small angles to those of the rolling movement (cf. *Organ*, 1931, No. 19). The effect of movements so damped out and the question of knowing if they degenerate into hunting properly speaking depend upon the state of the following sections of track. All these movements are controlled, moderated, and damped out by the centring springs.

The bogies consisting of one pair of carrying wheels and one pair of driving wheels are only found, on the steam locomotives dealt with in this report, on the 2-8-2 tender locomotive, series 19°, of the German State Railways; on the other hand, they are much more numerous on electric locomotives, for the reasons given in Chapter I. As regards running through curves, and guiding, it is not a matter of indifference if the driving pair of wheels is mounted with the pair of carrying wheels in front of it in a common frame, as in the case of the Java bogie, or if it is placed, as on the Krauss-Helmholtz bogie, on the main frame of the locomotive and cannot then move except transversely. In the same way, it matters whether the centre is located between the two axles of the bogie or behind the coupled axle.

We have explained in sufficient detail in Chapter II, when speaking of the E 17 locomotive, *how the Krauss-Helmholtz type of bogie runs through curves.* It is nonetheless useful to add a few words, as one of the most recent works on electric locomotives for main lines says in this connection that « a deviation of the leading carrying pair of wheels towards one side causes geometrically a parallel and simultaneous displacement of the driving pair of wheels towards the other side ». A statement of this kind does not satisfy the actual facts. As early as 1906 (cf. *Zeitschrift des Vereins deutscher Ingenieure*, 1906, page 1553), when explaining a similar fact, Helmholtz himself pointed out how much expressions of this kind on the action of forces on the moment of running onto

the curve give rise to incorrect ideas, ending in the conclusion that the Krauss-Helmholtz bogie is much less satisfactory than the bogie consisting of two pairs of carrying wheels. They produce the impression, wrote Helmholtz, that the principal factors are the carrying pairs of wheels and the centre, and that the coupled pair of wheels should behave as required by the pair of carrying wheels, or in other terms that the coupled pair of wheels is suddenly thrown over to one side at the moment of entering the curve. Cause and effect are confounded. The only decisive factors are the two pairs of wheels and it is the centre which has to obey. There is not therefore — we ourselves add — any difference in principle as compared with guiding by a bogie with two pairs of carrying wheels with its two outside wheels bearing against the directing rail. This is moreover the explanation why this bogie gives equal satisfaction under steam locomotives and electric locomotives, after having been fitted with transverse spring gear of the usual type (two laminated springs placed one behind the other).

It is necessary to be fully conversant with these details when it is a question of judging between the types of bogies. In the Kleinow type of the Krauss-Helmholz bogie (cf. locomotive E., 17, Chapter II), transverse movement of the frame can easily take place, even when the pair of driving wheels is braked, because it is not necessary here to overcome any transverse friction between the slide and the axle box guide. Consequently the horns which gripe the driving axle box as between shears can be built relatively light.

We have already, in Chapter II discussed the Java bogie, of the Buchli type. The description of the 1 Do 1 locomotives led us quite naturally to point out the differences between the Krauss-Helmholtz and the Java types.

It may be asked why the results obtained with the Java bogie, by the two railways mentioned, *did not in any way agree.* A reply must be given to this question as it affects the guiding in its most essential point. The following is our opinion on this matter :

With the Krauss-Helmholtz bogie the two outside wheels of the carrying pair and the coupled pair undoubtedly come into contact with the outer line of rails at certain angles. The guiding effort of the centre is transmitted, as in the case of the bogie with four pairs of carrying wheels, through the centring springs. There is no unsprung connection with the main frame of the locomotive. The position is quite different in the case of the Java bogie. The coupled pair of wheels is mounted on the bogie frame. The point of support for the change of direction is situated behind the pair of driving wheels in the main frame. All the irregularities of the track, both in height and direction, act directly upon the body of the locomotive (E 16) with its electric equipment, owing to the rigid centre fitted on the main frame. The intermediate elastic detail which is provided on the Krauss-Helmholtz bogie is missing. To this must be added that the resultant of the tractive effort of the coupled pair of wheels of the bogie and of the resistance the main mass of the locomotive opposes against the leading bogie endeavouring to advance, tends to increase on a curve the striking angle of the leading guiding pair of wheels. To this tendency to derailment the coupled pair of wheels can only offer little resistance at the moment that this resultant force is set up, i. e. by slight wedging and a striking angle that is still small, because the coefficients of friction still have only a low value.

Consequently the Java bogie easily finds itself, when its leading guiding pair of wheels makes contact, in an unstable state of equilibrium. The tendency to increase the striking angle of

the guiding wheel is, moreover, independent of the position of the centre, which may be behind, or between the axles of the bogie, when the coupled pair of wheels and the carrying pair of wheels are mounted in a common frame. Consequently the reflexions given above would also be applicable, for example, to the Java type of bogie.

If, therefore, the Dutch East Indies State Railways and also, as we learnt when closing this report, the Swiss Federal Railways are very satisfied with their Buchli bogies, this is undoubtedly due to the fact that they have been able to remedy the features that we have just pointed out, either by a guided length which is considerable relatively to the rigid wheel base, or by strong centring springs acting on the great lever arm placed above the carrying pair of wheels.

If we now compare the two 1 Do 1 locomotives, we find that the guided length of the Dutch East Indies State locomotive is equal to 2.99 times its rigid wheel base and that the guided length of the E 16 locomotive of the German State Railways is only 1.9 times its rigid wheel base. These are very important differences in favour of the Dutch Indies Railways' locomotive. It appears, moreover, that the Java bogie of the German 1 Do 1, series E 16 locomotive also gave less satisfaction than the Krauss-Helmholtz bogie, because the centring device which had been fitted slightly behind the transverse centre line of the bogie was much less able to overcome hunting movements than the pair of laminated springs fitted on the 1 Do 1 locomotive of the Dutch East Indies State Railways above the leading carrying pair of wheels of the bogie.

Individual pairs of wheels have not been used except in quite isolated cases as guiding pairs of wheels on high-speed locomotives; naturally these are pairs of wheels of the Bissel or Adams type which can readily run through curves; we find them more frequently on the electric lomotives considered in this report. We have broadly sketched in Chapter I what we think of the aptitude of these pairs of wheels to guide vehicles at high speeds. As they are not fitted with centring springs of sufficient force, these pairs of wheels do not guide at all; the carrying pairs of wheels fitted with centring device are not suitable for vehicles intended to run at high speeds except in rare cases. Experience confirms this remark. The German State Railways' Central Designs Office has reported unfavourably on the running of the 1 Bo-Bo 1, series E 18 locomotive fitted with a pony truck. The Swiss Federal Railways have found that in the case of their 2 Do 1 locomotives, which are guided in one direction by pony trucks and in the other by Java type bogies, that, when guided by the bogie, the vehicle runs much more steadily than when the pony truck guides, and at the same time the wear of the flanges is much less.

After what has been said above on the value of Y : Q, we now clearly see how necessary it is to give the greatest possible length to the lever arm of the guiding force of the centre, which has to assure the rotation of the vehicle about its centre. The distribution of the guiding between the bogie and a leading coupled pair of wheels rigidly mounted in the frame is incompatible with this desideratum. We have, moreover, mentioned in Chapter I that the accidental participation in the guiding of the coupled wheels rigidly carried in the frame is unavoidable.

The transverse play of the coupled wheels fitted in the rigid wheel base — an arrangement which Helmholtz showed the usefulness of by calculation (*Zeitschrift des Vereins deutscher Ingenieure*, 1888) and Gölsdorf by the practical test carried out about 1900 — *is*

the second method of reducing the turning force and consequently the guiding pressure Y. We know that the transverse play should be sufficiently great so that when the vehicle is running through a curve the pairs of wheels exercice no transverse pressure on the main frame of the locomotive, i. e. that each individual pair of wheels should be guided directly by the outer rail. The guiding pressure has therefore nothing to do except deal with the side forces, the direction of which depends upon the rail, produced by the forces of friction which exist at the points of support of the wheel. Undoubtedly the thinning of the flanges or the suppression of the flanges — a method used in very rare cases — cannot in any way contribute to reducing the guiding pressure. Among the electric locomotives with individual axle drive, mentioned in the replies, there are a great number in which the coupled pairs of wheels have only side play.

Among other external forces, the action due under certain conditions to *the coupling between the locomotive and the tender* exercises an influence on the guiding pressure at the leading guiding wheel of the locomotive. On the German State Railways the locomotive and the tender of the standard type are connected by a rigid coupling which absorbs the tractive and rebound effort of the locomotive. Auxiliary couplings are provided to come into action in the event of the main coupling breaking. The form of coupling used by the Danish State Railways and on part of the vehicles of the North of Spain Railways are alike, except that the auxiliary rigid couplings have been replaced by safety chains. Contrary to our expectations, we have still found couplings of the carriage and wagon type on a small number of engines. These couplings can in no way help to damp out the rebound movement. Generally small buffers are fitted on both sides of these main couplings and serve to damp out hunting movements. These buffers bear with slightly curved heads against the buffer plates fitted on the back buffer beam of the locomotive, or they have wedge shaped heads (German State Railways, Finland, Holland), which bear against inclined guides, and offer considerable opposition to hunting movements, but on the other hand, as Dr.-Ing. Heumann has shown in the *Organ*, 1918, increase the guiding pressure to an appreciable extent. The Danish State Railways and the North of Spain Railways use, on part of their rolling stock, buffers the centres through which cross on the main draw gear. The Norwegian 4-6-0 locomotive, which is coupled to the tender in the usual manner by a rigid drawbar and two safety couplings, has no spring buffer between the locomotive and the tender. The tension is got by means of a plate tapered off to each end bearing against a flat plate; the back of the double wedge is on the transverse axis of the vehicle and is fixed against the flat buffer plate. Adjusting devices prevent the setting up of longitudinal play. The 2-8-2, series 4500 locomotive of the North of Spain Railways has a similar centre coupling acting along the longitudinal centre line of the vehicle; the drawings do not show if it is fitted with springs or not. Little information has been supplied upon the individual tension of the springs of buffers between engine and tender; it is 1 800 to 2 000 kgr. (3 960 to 4 400 lb.) for buffers bearing against flat buffing plates, and 8 000 to 18 000 kgr. (17 600 to 39 600 lb.) in the case of buffers in the form of wedges. For the standard locomotives of the German State Railways, very strong buffer springs are used on the tenders (on the tenders 4 T 32, for example, the final tension is 21 000 kgr. [46 200 lb.]), in order to make absolutely certain that the mass of the tender will damp out the rebound. The buffers are connected

together by equalisers with the very important object of assuring equal pressure on the buffers when running through curves.

On electric locomotives, neither rebound nor rolling movements are experienced. Nonetheless it is necessary to study carefully *the connection of the motor bogies of electric locomotives with one another* and with the body, from the point of view of its reactions on the guiding pressure. The information taken from the replies or collected in other ways makes it possible to say the following on this subject : the body of the 1 Bo-Bo 1, series E 18 locomotive of the German State Railways is not subjected to any traction effort; the two motor bogies are joined together by a connecting bar, but are not in any other way held together. The motor bogies of the 1 Co-Co 1, series 7 100 locomotive of the North of Spain Railways are not connected together nor held against one another, while the motor bogies of the Co-Co locomotive series 7 000 of the same company are held together by means of guides in the form of wedges, and those of the 2 Co-Co 2 locomotive, series 7 200, are connected by a draw bar with springs and protected against the effects of hunting by a coupling bar fitted with springs, arranged at an angle to the longitudinal centre line of the locomotive.

The greater the ratio of the guided length (cf. Chapter I) to the rigid wheel base, the smaller, as a rule, is the force turning the main frame and the smaller as a whole is the necessary effort at the leading end for guiding. *It is desirable therefore that the guided length selected should be proportionally as long as possible.* To give some idea with figures of the values of the directing force and of the guiding pressure, when the guided length varies, we have studied, using the method indicated by Heumann (*Organ* 1913 and 1928), the guiding conditions of the 4-6-0, series $17^{10\text{-}12}$ locomotive of the German State Railways.

We first of all decided for this locomotive on a guided length of 8 m. (26 ft. 3 in.), the centre of friction, i. e. the common centre of rotation for all the frictional forces between the wheels and the rails, in the case in which the main frame of the locomotive freely advances and in which consequently the driving and coupled wheels do not come into contact with the outer rail nor the inside rail on a curve. Thereafter we have ascertained the guiding pressures corresponding to the leading wheel of the bogie, when the main frame makes contact with the outer rail and the inner rail on curves of different radii. The greatest guiding pressure occurs when running free and is equal ta $2.386\,\mu\,Q$ (μ being the coefficient of friction and Q the average pressure of the driving wheels) in the case with which we are dealing, with 23 mm. (29/32 inch) lateral play between the pair of wheels and the track and on curves of 303 to 350 m. (15.015 to 17 1/2 chains) radius. On greater curves the first rigid pair of wheels takes its part in the guiding; if the radius is 1 510 m. (75.5 chains), the guiding pressure on the guiding wheel of the bogie falls to $1.823\,\mu\,Q$. In curves of small radius the fifth pair of wheels comes in contact with the inside rail, and here again the guiding pressure of the leading wheel again falls slightly although in an insignificant proportion.

Now to show the influence of a greater or smaller guided length, the bogie centre has been moved 1 m. (3 ft. 3 3/8 in.) forward under the hypothesis that on moving the boiler it is possible to obtain the same axle loads as before. In this way the guided length has been increased to 8 + 1 = 9 m. (29 ft. 6 3/8 in.). With the same object again the guided length has been reduced to 7.40 m. (24 ft. 3 23/64 in.) by bringing back the centre as far as the design would allow. The investigations carried

out either on the drawing board or by calculation gave the following results :

Thanks to a long guided length it is possible for the locomotive to run freely, i. e. not to be guided except by the centre on curves of a relatively large radius varying between wide limits. Whereas with a guided length of 7.40 m. (24 ft. 3 23/64 in.) free running is only possible on curves of between 252 and 254.5 m. (12.6 and 12.775 chains) radius, the limits of the radii of curves increase from 303 to 350 m. (15.015 to 17 1/2 chains) for a guided length of 8 m. (26 ft. 3 in.), 413 to 567 m. (20.65 to 28.35 chains) for that of 9 m. (29 ft. 6 3/8 in.), and 585 to 904 m. (29.25 to 45.2 chains) for that of 10 m. (32 ft. 9 3/4).

The turning effort P_d of the centre and the guiding pressure Y have smaller maximum values when the guided length is considerable; consequently the safety against derailment increases. Table 11 below shows to what extent the maximum values of the turning effort and the guiding pressure vary when the guided length is increased or reduced. It also shows the improvement or the lowering of the values P_d and Y relatively to those of the guided length of 8 m. actually existing.

TABLE 11.

For a guided length of	P_d / μ Q		Y / μ Q	
	becomes	+ grows worse — improves %	becomes	+ grows worse — improves %
7.40 m. (24 ft. 3 23/64 in.)	2.125	+ 9.5	2.462	+ 3.18
8.00 m. (26 ft. 3 in./	1.94	0	2.386	0
9.00 m. (29 ft. 6 3/8 in.)	1.682	— 13.3	2.224	— 6.8
10.00 m. (32 ft. 9 3/4 in.)	1.47	— 29.4	2.152	— 9.8

Table 11 only shows the amount the maximum values P_d and Y corresponding to the free running of the main frame are made worse or better. The curves of figure 28 show how the turning efforts and the guiding pressures vary, for given curves, with the guided lengths of different values. It is surprising to see that in the case of curves of large radii, the turning effort and guiding forces are in reality slightly higher for locomotives with long guided lengths than for locomotives with short guided lengths. The differences are however of little importance. In our example the avantage of the long guided lengths is only obvious in the case of curves of less than 500 m. (25 chains) radius. Here the turning efforts and guiding efforts of the locomotive exceed the value for the locomotive with long guided length and reach, by increasing rapidly, the maximum values quoted in table 11.

Between the guiding bogie with lateral play and the centre to which the turning effort is transmitted, an intermediate part must be placed. This part is the *centring device.* According to the manner in which the centring force acts, there are centring devices with springs and centring devices without. The two kinds are represented on the locomotives

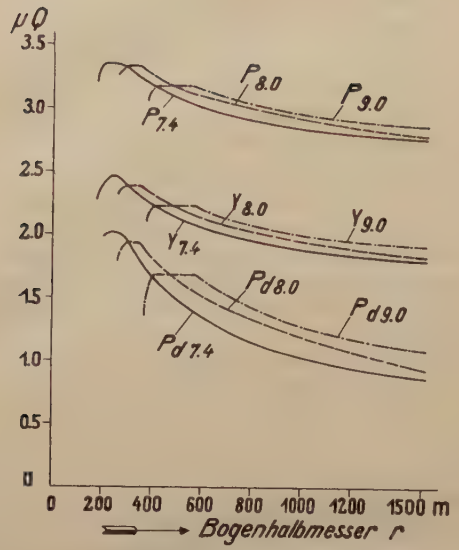


Fig. 28. — Influence of the guided length on the centring effort P*d*, turning force (P) and guiding effort on a 4-6-0 locomotive.

mentioned in the replies. *We will consider first of all the arrangements with springs.*

We have already explained, when speaking of the running gear at the leading end, how important it was that the inevitable transverse shocks should be absorbed elastically, and that every method should be employed to guide the vehicle elastically. We will recall that the expression « centring device » only represents part of the work to be done by such fittings. Laminated springs are better for this purpose than helical springs. This superiority lies in the ability of laminated springs to damp out more quickly all vibrations and all shocks. Certain part of the pressure of the guiding wheel flange corresponds with the centring force. For this reason there are limits to the value of the centring force. If the initial tension is very small, states Uebelacker, the vehicle runs quite straight during a relatively great part of the time; the striking angle and the side movement have to be very great before the centring force comes into action appreciably; its subsequent values will be proportionally higher. *Starting from the initial tension the tension of the centring devices increases linearly, but owing to the pressure of the flanges it is necessary that the increase of the tension should remain small.* The builder can make the final tension any value desired in the simplest way; alterations subsequently desired are easy to carry out. It is otherwise in the case of bolsters. Their initial tension is nil, graduation of the centring forces is not possible, that is without considerable alterations to the spring gear.

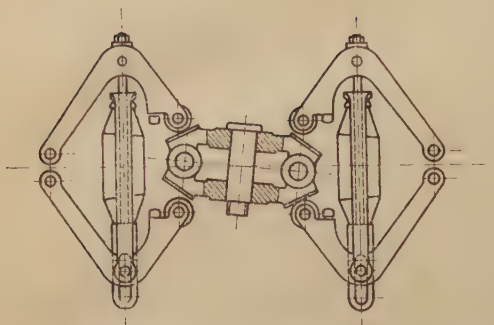
This is why the development of these details during the last twenty years has evolved in the sense indicated above, which is to only let the centring forces of the elastic centring devices increase with the moment of the centre progressively. Bogies of the Krauss-Helmholtz type were formerly fitted with a single laminated spring, the centring force of which was proportional to the total value of the deviation. Simple centring springs have been replaced on these bogies as well by two laminated springs in series arranged in the well-known manner on both sides of the centre and connected together by tie rods. Most of the bogies the designs of which have been sent to us are fitted with this centring device. The tension of the springs increases porportionally to half the value of the deviation of the centre because each of these two springs takes the same deflection under the turning force. The travel is doubled. Conversely the same travel would correspond in the case of a single spring to a force twice as great. On all the bogies, i. e. on all the four-wheeled carrying bogies as well as on the bogies of the Krauss-Helmholtz type and its modifications, the centring force is as a rule only applied at the centre. In this case the damping out of the hunting movement only takes place after the

bogie has made a slight transverse movement relatively to the centre. This is why on the Kleinow bogie (variation of the Krauss-Helmholtz) of the 1 Do 1, series E 17 locomotive in addition to the usual centring springs placed in line with the centre (initial tension 2 180 kgr. [4 880 lb.], final tension 2 375 kgr. [5 320 lb.], a second centring device has been fitted above the leading carrying pair of wheels (initial tension 550 kgr. [1 230 lb.], final tension 1 550 kgr. [3 470 lb.]) which is able entirely, because of its position, to overcome the hunting movements at the actual moment at which they arise and moreover does this duty successfully. One of the reasons why the Java bogie of the Dutch East Indies State Railways, fully dealt with in Chapter II and in Chapter IV, has given full satisfaction, whereas this was not the case when experimentally tested on the German State Railways, has been the use of the special centring device above the leading carrying pair of wheels.

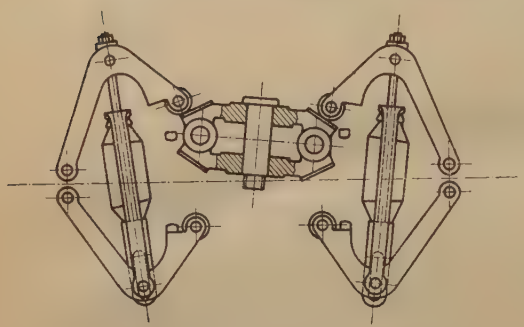
The endeavours made by modern locomotive builders to observe the proper proportion between the initial and the final tension of the centring device is shown most clearly in the Winterthur type bogie used on the 2 Co-Co 2 locomotives of the North of Spain Railways. Bogies of this type have been built for the first time in 1924 for the Paris-Orleans Railway.

Figure 16 shows the method of construction of this bogie and its centring device. The frame of the locomotive is carried on two slides *g* by means of the articulated roller *z* and the two hemispherical roller supports *k* connected to the articulation of the first. In this way the frame is supported on a transverse knife edge and a longitudinal knife edge, i. e. at the centre of the bogie. The centring forces are exerted by the rollers *r* on the articulated detail *z*. These rollers act on the free end of the levers turning about the fixed points D. The lever arm of the springs varies with the travel of the articulated detail. Its maximum is reached when the longitudinal centre line of the bogie and the longitudinal centre line of the locomotive coincide. Each rotation of the longitudinal centre line of the bogie about the ideal centre sets up a moment of rotation behind the two pressure rollers *r* diagonally opposed. When the bogie is moved laterally the moment of rotation backwards is smaller. Figure 29 shows under the heading « Rotation of the bogie without lateral movement » the conditions of simple rotation (hunting) and under the heading « Position on curves of small radius » the condition of the bogie at the end of its travel.

The North of Spain Railways have not supplied any information on the value of the centring forces. It is only possible to judge of the practical utility of this type by means of numerical data or from graphs. The builder of these bogies informed us that the initial and final centring forces of the Spanish locomotive were only slightly less than the centring forces of the bogies of the Paris-Orleans Railway, in which the main initial force is 4 500 kgr. (10 000 lb.) and the final, with 163 mm. (6 7/16 inches) lateral displacement, 5 900 kgr. (13 210 lb.). By means of these values we have drawn the curves of the centring force against the lateral displacement (*s*), of the spring force against the simple rotation without displacement (*v*) and of the moment of rotation in the reverse direction against the simple rotation (*m*), and we have then shown them in figure 29 above the lateral movement and the angle of rotation. The arrangement of the levers results in the final centring force becoming 1 650 kgr. (3 690 lb.), or 28 % less than it would have been had the centring force increased directly with the lateral movement, in the same proportion as the tension of the springs (curve *t*), a case which would occur



Rotation of the bogie without lateral movement.



Position on curves of small radius.

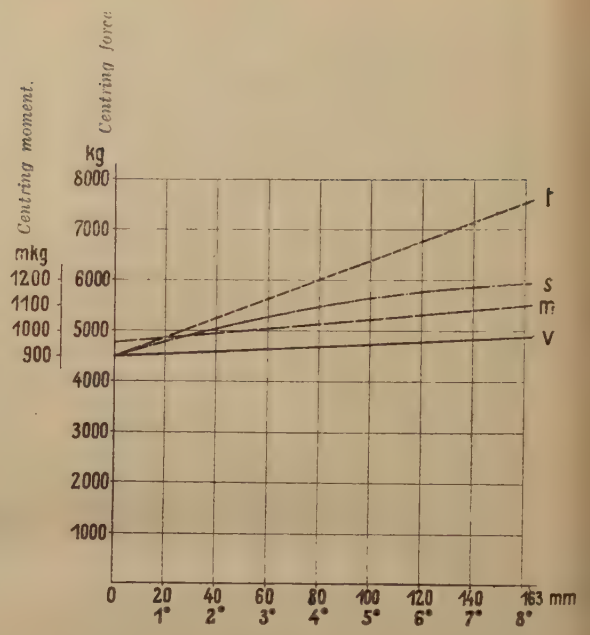


Fig. 29. — Winterthur type centring device.

with other spring centring devices. By a proper selection of the ratio between the initial tension and the final tension of the springs and by a suitable arrangement of the levers it should be possible to keep the centring effort constant or even make it decrease with an increasing side movement.

Against the rotation without lateral movement, between 1° and 3° of rotation, there is a centring force of 4558 to 4 628 kgr. (10 200 to 10 360 lb.) exercised by the two rollers arranged diagonally relatively to one another and a moment of rotation in the backward direction of 1 000 kilogrammetres (7 200 foot-pounds) in round numbers. The centring force against the rotation and the moment of rotation in the backward direction increase directly with the angle of rotation.

The curve *s* also applies approximately to the case when there is simultaneously lateral displacement and slight rotation. Strictly speaking, the centring force in this case is less, but the difference is not more than 2.2 % in the extreme position, for an angle of rotation of 1°. It is true that the moment of rotation in the backward direction is then exceedingly small, because the difference of the moments of pressure of the rollers on the side of the lateral deviation is now all that there is to cause rotation in the backward direction. The small reverse moment of ro-

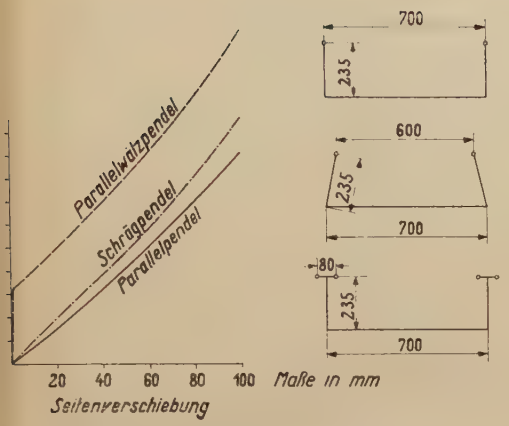


Fig. 30. — Centring for curves.

Explanation of German terms:

lelwälzpendel = Parallel links with rollers. = Schrägpendel Inclined links. — Seitenverschiebung = Side movement.

tation is aided in its action of damping out hunting by the internal friction of the springs and joints of the levers, so that even in this case this type is better than the usual types.

The support of the bogie and the centring device of the Winterthur type represents a form of construction worthy of attention. It is true that the advantages obtained from it have to be paid for by a rather complicated design.

The centring devices based upon the action of gravity — we will only consider here *bolsters — are less suitable in our opinion for high speed locomotives* and this chiefly on account of their lack of capacity for the elastic absorption of the transverse forces. A certain number of the locomotives mentioned in the replies are fitted with *such bolsters, with either parallel links, inclined links, or triangular links* (parallel links with rollers). In the central position, the initial centring force of all these bolsters is zero. This is their second defect when considered from the point of view of our question. From this centre position the centring force increases more or less quickly. Our investigations into the running on curves shown by the values of the centring forces above the lateral deviations end in the following conclusions (fig. 30) : With equally long links and equal horizontal spacing of the lower points of suspension of the bolster, the centring force increases quicker with inclined links (inclination of 1 in 4.592 as in the case of the bolsters of the old Saxon locomotives) than with parallel links. With triangular links of the same length and an average distance apart of 80 mm. (3 5/32 inches) of the links, the smallest oscillations are sufficient to cause the centring force to increase rapidly. Figure 30 shows how, from the above, the centring forces correspond to the increase of the lateral movement. The value of the centring forces depends with inclined links upon the length of the links, their inclination and the horizontal and the distance apart of their lower supports (short links inclined at a great angle give large centring forces); with the parallel links it depends upon the length of the links, and with triangular links on the length of the links and the distance apart of the bolts of each of the roller supports.

These bolsters are used on the bogies with carrying pairs of wheels as well as with pony trucks. Ten of the steam locomotives mentioned in the replies use the different types. Bolsters with inclined links are found on four locomotives : the old 4-6-0 and 4-6-2 Saxon locomotives, the 4-6-0 locomotive of the Saxon State Railways and the 4-8-2 North of Spain locomotive. Triangular links are used on five locomotives : the two 4-6-0 locomotives of the Finnish State Railways, the 4-6-2 locomotive of the Swedish State Railways, the 2-8-2 of the North of Spain Railways and the 4-8-2 locomotive of the Madrid-Saragossa Alicante Railways. Parallel links are only found on one locomotive, the 4-6-0 of the Bergslagen Railway. Not a single

one of the electric locomotives dealt with in the present report is fitted with centring devices using a bolster.

What centring force should be provided with centring devices and what, from this point of view, does the information used to compile this report teach us ? If we remember that the static guiding pressure which holds the guiding wheel with its flange against the outside rail is due to the other wheels not in contact, that it is necessary to take into account the guided length and the details of the design of the bogie, and that it is not a matter of difference whether the guiding takes place at two points or at one alone, the answer is easy; it should be as follows : *the centring force should be at least equal in value to the turning force acting at the centre.* It is also necessary that it should be able to deal with the transverse shocks to which the leading wheel is subjected and that it should therefore be greater than that required by purely static conditions. Here again the maximum value of the ratio Y : Q can supply the criterion for safety against derailment. And it is there, from the detailed explanations we have given above on the subject, that the difficulty lies of giving an exact reply. The running of the bogies never being in a permanent static state of rest, but being constantly sinuous and able to degenerate through fortuitous circumstances into hunting, it is necessary to stress in this order of ideas the necessity for an elastic centring device, able to absorb rapidly the vibrations which are set up. It is for this reason that, from the point of view of Question III, helical springs used in the leading bogies of the steam locomotives of the Netherlands Railways and on the North of Spain Railways on steam and electric locomotives cannot be recommended. The other pairs of wheels are often fitted with coiled springs; there is no objection to their use in this position. The information with which we have been supplied as to the value of the centring forces corresponds with the diversity of the different locomotives regarded as vehicles and the different ideas of the administrations as regards guiding at one or several points. This will be shown by a few figures.

The initial and final tensions of the *centring springs of the bogies of steam locomotives* lie between :

1220 and 5 500 kgr. (2 690 and 12 100 lb.) for the 4-6-0 locomotive of the Norwegian State Railways.

2 000 and 3 240 kgr. (4 400 and 7 140 lb.) on the 2-8-2, series 19 locomotive of the German State Railways.

1 700 and 4 300 kgr. (3 740 and 9 470 lb.) on the standard locomotive, series 01, etc., of the German Railways.

In the case of the *bolsters of the bogies of steam locomotives,* the initial and final tensions lie between :

0 and 2 200 kgr. (4 840 lb.) (inclined links) on the 4-6-0 locomotive of the former Saxon State Railways (fig. 17).

0 and 4200 kgr. (9 240) (vertical links) on the 4-6-0 locomotive of the Bergslagen Railway.

2240 and 6 820 kgr. (4 930 and 15 000 lb.) (triangular links) on the 4-8-2 locomotive of the Madrid-Saragossa-Alicante Railways.

1 950 and 4 900 kgr. (4 290 and 10 790 lb.) (triangular links) on the 4-6-2 locomotive of the Swedish State Railways.

The whole of the electric locomotives are fitted with centring devices using springs; the initial and final tensions lie between:

350 and 1 000 kgr. (770 and 2 200 lb.) on the 2 BB 2, series E 52 locomotive of the German State Railways.

1 000 and 6 400 kgr. (2 200 and 14 100 lb.) on the 2 C 2, series E 66 locomotive of the German State Railways.

1 200 and 1 650 kgr. (2 640 and 3 630 lb.) on the 1 Do 1, series E 16 locomotive of the German State Railways.

2 160 and 2 375 kgr. (4 760 and 5 230 lb.) on the 1 Do 1, series E 17 locomotive of the German State Railways.

On the last mentioned locomotive a spring with additional plates is fitted above the leading carrying pair of wheels, the initial and final tensions of which are 550 to 1 555 kgr. (1 212 to 3 427 lb.).

In the case of *the guiding pairs of carrying wheels* upon which we have received information the initial and final tensions lie between :

870 and 2 750 kgr. (1 910 and 6 060 lb.) on the 1 Do 1, series 16^5 locomotive of the German State Railways;

1 755 and 2 590 kgr. (3 870 and 5 700 lb.) on the 1Bo-Bo1, series E 18 locomotive of the German State Railways;

1 750 and 3 500 kgr. (3 860 and 7 920 lb.) on the 2 Do 1, series E 21 locomotive of the German State Railways.

Among the *trailing pairs of carrying wheels,* those fitted with centring gear have springs with the following initial and final tensions :

between 680 and 3 800 kgr. (1 500 and 8 370 lb.) on the standard locomotives of the German State Railways;

between 1 250 and 2 950 kgr. (2 750 and 6 500 lb.) on the 2 C 1, series 19^5 locomotive of the German State Railawys;

between 1 030 and 1 370 kgr. (2 270 and 3 020 lb.) on the 2 D 1 locomotive of the Madrid-Saragossa-Alicante Railways.

As we have said on several occasions, the value of the static turning force is calculated from the equation of equilibrium of the external forces applied to the vehicle; amongst these are first of all the frictional forces between the wheel and the rail, which oppose the rotation about the centre or centres of friction. The more accurate determination of the coefficient of friction used in these calculations was the subject of investigations by the technical committee of the German Railway Association, enquiries which had not yet been completed at the time of drawing up this report. The results ascertained in 1931 are given in No. 19 of the *Organ,* 1931. We learn from this publication that under the given outside conditions the coefficient of friction of the transverse sliding movement of the revolving *wheels other than the driving wheels* depend only upon the ratio of the speed of running to the speed of sliding and of the degree of stressing of the surfaces in contact, that these coefficients of friction are very small for small angles and hardly reach the value of the coefficients of friction of the simple sliding movement at about 1°. We see moreover that the coefficients of friction of the rolling and sliding movement of the driving and coupled wheels are still smaller. Here we shall be satisfied with calling attention to the remarkable progress made in arriving at the values of Y. We now know how it happens that the wheels of the bogies so easily hunt. The slightest deviation in direction of the track (for example, small changes of gauge of the track) are equivalent to a constraining force. Against the transverse movement there is a resistance which is about equal in strength to the resistance to simple rolling movement. The results show moreover, as regards Question III, Part I, how small angles of constraint suffice to keep Y to a low value.

Safety against derailment increases if care is taken to see that the friction at the point of contact of the flange of the guiding wheel remains as low as possible. This consideration is only realisable by lubricating the flanges or, a less usual method, by lubricating the sides of the rail heads on curves of small radius. Very varied solutions have been tried in this matter.

Out of the ten administrations consulted, five lubricate the flanges; two of

these railways, the Dutch East Indies State Railways and the Bergslagen Railway, have given up the tests, as the results were not satisfactory. Consequently three administrations alone are continuing the tests : these are the German State Railways, the Norwegian State Railways and the North of Spain Railways.

The most simple method of lubricating the flanges of steam locomotives consists in directing against the guiding flanges exhaust steam from the steam water heaters and the air pumps, as it contains oil. On the locomotives of the former Baden and Saxon State Railways good results have been obtained with this arrangement, although the amount of oil in this exhaust steam is rather small. The German State Railways are not satisfied with these results and at the present time are considering how they could be improved by other means. They are carrying out tests of the following devices :

The greaser of the type used on the Paris-Orleans Railway, which automatically directs, as needed, pulverised oil by means of compressed air onto the curve on the top inner edge of the left hand or right hand rail of the curve;

lubrication by grease on the German State Railways method; in this case the flange is lubricated. A wood tube pressed against the flange by a spring is filled with solid grease;

a Rössger type oil lubricator, in which the oil is drawn in by compressed air and sprayed against the root of the flange; and

a Heuser type oil lubricator, which feeds the oil to the flange by means of a steel ball.

Three other types of lubricator, Limon Flume, Bosch and Hasler, tested by the German State Railways have not given satisfactory results. All these arrangements undoubtedly find it difficult to meet the restriction that they must lubricate, and that too very lightly, only the zone of contact between the rail and the flange.

On some locomotives of the Norwegian State Railways, the guiding flanges are lubricated by a brush pressed against the root of the flange, the other end of the brush being in an oil container. The inside edge of the rails of one particularly sharp curve is also hand oiled.

The North of Spain Railways lubricate, successfully, the flanges of certain electric locomotives by means of a simple lubricator, and by means of a Bouclon type lubricator. They have given no other information on the subject.

The problem of the effective lubrication of the flanges of the guiding wheels does not yet appear to have been solved in a satisfactory manner by all administrations. In the spirit of Question III, first part, attention should be called to the usefulness of providing a solution in the near future.

The question of knowing in what proportion the inevitable irregularities of the track modify the vertical pressures and horizontal pressures during the running of the vehicle, is difficult to examine by calculation and has not yet been elucidated by trials. We have not received on this subject any useful information. The administrations limit their investigations to checking the static security against derailment. We do not mean to say in any way that tests have not been made to obtain, by approximate calculations, an idea of the magnitude of the dynamic actions.

The difficulties in the way of carrying out precise investigations lie in the determination of the masses concerned and the extent of the deformations of the track and the vehicle. How do these facts take place ? Undoubtedly as follows :

The movement of rotation of the vehicle in a curve must be set up on run-

ning into the curve and neutralised on leaving it. By means of transitions which are generally of the cubic parabola form, we can follow the processus of the angular acceleration which has to be produced and the lineal acceleration of the centre of the mass, transversely to the direction of running. In a way a certain duration is guaranteed to this processus. When all the wheels of the vehicle are on the curve, the accelerations have become equal to zero. These facts take place in the reverse order when suppressing the uniform movements of rotation. In the case of irregularities of the track on curves and straight alignments, but especially in the case of the sudden transition from the straight alignment to the curve, the required accelerations must be produced, but here the track requires that the angular accelerations and the transverse accelerations take place in a very short time and at very high values. The result of this naturally is a shock. It is therefore these places which can compromise the certainty of guiding. It the vehicle is fitted with elastic centring gear, the deformations (which constitute one of the groups of difficulties) of the rail, of the wheel, and of the frame, are small when compared to that of the spring centring gear.

Recently, however, the determination by calculation of the dynamic influences has made a step forward. We are indebted to Reichsbahnoberrat Dauner and Dr.-Ing. Uebelacker for this progress. We have already had occasion to quote the article by Dr. Uebelacker entitled « Über die Massenwirkungen bei plötzlichen Richtungsänderungen im Lauf von Eisenbahnfahrzeugen » (The action of the masses in the event of sudden changes of direction in the running of railway vehicles), *Organ*, 1930, page 271); this has been very useful to us in studying the centring gear. Dr. Uebelacker examines the case of guiding by a spring centring device, neglecting serious deformations of the rail, of the wheel, and of the frame, on a strictly mathematical basis, and shows the great importance that must be attached to elastic guiding and to the presence of transitions. If there are transitions on entering and on leaving curves, Dr. Uebelacker considers that the forces set up are insignificant.

In his article of April 1930, Dauner gives the turning force as the total of the guiding pressure due to friction and of the guiding pressure of the mass and from them combines the value for the striking angles of the guiding track of 1°, 1 1/2°, and 2°, with the speed. Measuring the guiding pressures according to the wheel pressures, which vary with the guiding pressures of the mass, and supposing that the downward movement of the guiding wheel continues to exist when $Y : Q = 1$, he finds the critical speed of the vehicle considered, for the different striking angles. For the « reduced mass » M_{red}, i. e. the part of the vehicle that can be imagined concentrated at the striking point, so that the action of shock is the same as on the actual vehicle, Dauner deduces the relation :

$$M_{red} = \frac{M}{1 + \frac{d^2}{\rho_d^2} + \frac{w^2}{\rho_w^2}},$$

in which:

M is the total mass of the vehicle,

d the horizontal distance from the striking point of the guiding wheel to the vertical transverse plane through the centre of gravity,

w the vertical distance of the point of contact to the horizontal plane passing through the centre of gravity,

ρ the radius of inertia existing in the directions of d and w.

The guiding pressure of the mass according to Dauner is :

$$P_m = v \sin \alpha \sqrt{\frac{M_{red}}{c}},$$

in which :

v is the speed,

α the striking angle of the guiding wheel,

c the amount of the flexibility of the wheel and the rail at the striking point, or the coefficient of elasticity.

The results given by these equations agree with what we have shown in the different passages of this report when considering Y and the guided length. Our thesis tending to assure the guiding at the front end as much as possible by a single turning force, i. e. not to let the driving wheels outside the bogie take any part in the guiding, finds its mathematical confirmation here. The equations show in fact that the magnitude of the « reduced mass » is a function of the position of the point of shock relatively to the centre of gravity. The term $\frac{w^2}{\rho^2 w}$ of the denominator indicates the influence of the up and down movement, taking into account the influence of the height of the centre of gravity of the engine as a whole above the top of the rail. Consequently the dynamic influences act both on the changes of guiding pressure and on the changes in wheel pressures.

The fact that the turning effort, whether of static or dynamic origin, imposes a supplementary vertical load on the wheel making contact with the rail and acts therefore towards improving the safety against derailment is known and taken into consideration by those who have to occupy themselves with this question. It is useless to call attention to it each time it is a question of Y. We have done so several times.

If it is a question of a locomotive without spring centring gear, the quantity c in the equation P_m of Dauner gives still rise to certain difficulties because we possess little sure information on the subject of c. We ought to collect data to be able to take full advantage of Dauner's investigation.

The work of a learned society in which we are interested suggested the idea of calculating, by measurements on the guiding wheel, the values of the loads on the wheel and of the guiding effort which actually exists on the running vehicle. Because of the exceptional importance of tests of this kind, the German State Railways agreed to a request made in this sense by the said society. The necessary apparatus has been prepared in the Berlin Higher Technical School in collaboration with the Heinrich Hertz Institute for investigations on vibrations. The tests were undertaken in the summer of 1932, and the interpretation of the oscillograms was in hand when this report was completed. The apparatus used were : instruments for measuring the displacements, accelerometers, and measuring devices which noted the changes occurring in the heights of the springs and in the vertical and horizontal loads of the guiding wheels of the locomotives and the tenders and marked them on the bands of the oscillographs. The object of these tests was first of all to ascertain to what point the more or less important irregularities in the track exercise unfavourable influences on the steady running of the vehicles and to determine numerically the effect of these conditions. It is also expected that the question of knowing if what gives the optima values statically — we will mention only the support characteristics C_m and C_v (cf. Chapters II and III) — still remains the optimum value from the dynamic point of view will be elucidated. As the bogies having the characteristice Ce and Cs were used on the test locomotives, it is to be hoped that it will be so.

We believe that we have in this way set out what is considered essential on the question of guiding in the present state of our knowledge on the subject.

The dynamics of the vibrations proper to steam locomotives and especially those which are due to the effects of the unbalanced reciprocating masses is a separate chapter on which we will only say here a few words, as we are dealing in this report with questions of guiding. It will soon be a hundred years since science began to take an interest in these vibrations; we may call to mind the names of Pambour and Redtenbacher. We also know by personal experience that vibrations of this kind can only become dangerous in the case of resonance, and we know the means of preventing this. Among the thirty steam locomotives considered in this report, there are :

10 two-cylinder, simple expansion locomotives;

13 compound four-cylinder locomotives;

3 four-cylinder, simple expansion locomotives; and

4 three-cylinder, simple expansion locomotives.

On all the two-cylinder locomotives, except 2, the reciprocating masses are balanced in the proportion of 18 % to 45 % by counterbalance weights The 2 two-cylinder locomotives with outside cylinders belong to the Dutch East Indies State Railways. We do not know the reason why no part of the reciprocating mass has been balanced.

In the case of 6 of the 13 compound locomotives with four cylinders, the reciprocating parts are not balanced : the same is the case for all the three-cylinder simple expansion locomotives.

On all the locomotives on which the reciprocating parts are partly balanced, the counterbalance weight is calculated in such a way that the free centrifugal force on each wheel at the highest speed of the trains shall in no case exceed 15 % of the weight on the wheel when standing.

To sum up the considerations given in chapter IV :

It is desirable that the locomotive should be guided by small turning efforts. The following ratio can be considered as criterion of immunity against derailment :

$$\frac{Y}{Q} = \frac{\sqrt{1 + \left(\frac{h}{l}\right)^2}}{\mu + \operatorname{cotan} \alpha}$$

in which Y represents the total guiding pressure and Q the weight of the guiding wheel (for the other terms, cf. Chapter IV, page 2315). The quantities h, l and α are dimensions as new; the information given by the replies to the questionnaire do not show to what point these quantities can be modified by wear of the tyres without compromising the safety against derailment. The dimensions given on this subject relate to the thickness and depth of the flanges. The North of Spain Railways alone report that α_{max} reaches 90°.

High-speed locomotives should be fitted at the leading end with bogies with two pairs of carrying wheels, or with bogies consisting of one pair of carrying wheels and one pair of driving wheels; but in accordance with the recommended practice of the German Railway Association, this latter type of bogie should only be used when, like in the Krauss-Helmholtz or similar types, the action of the tractive effort of the driving pair of wheels does not tend to increase the striking angle of the leading pair of wheels. The axis of rotation of these bogies should lie between the axles of the bogie.

With one exception, however, the Java type bogies, consisting of a carrying pair of wheels and a driving pair of wheels mounted in the main frame with its axis of rotation behind the driving pair of wheels, have given entire satisfaction.

The pressure of the flanges of the guiding wheels should be as small as possible. Therefore, the intermediate coupled pairs of wheels should have sufficient side play so that they do not exercise any side pressure on the frame of the locomotive.

Pony trucks are generally not satisfactory for guiding locomotives at high speeds ; when they are used special attention must be given to the centring device.

The guided length should be as great as possible. With a long guided length on curves of large radius the main frame is free; on the other hand, on curves of small radius high turning and guiding efforts are avoided.

High-speed locomotives should be fitted at the leading end with spring centring gear, the initial tension of which should not be widely different from the *final tension.* Considerable side play of the bogie may involve special designs (such as the Winterthur type). *Laminated springs are the most suitable for centring devices of the usual type. Among the centring devices using bolsters, those with parallel links with rollers are relatively the best for this purpose.*

Modern electric locomotives with a considerable total wheel base are sometimes guided, though indirectly, with as many as four driving pairs of wheels with lateral play. In these cases laminated springs must be used exclusively.

The considerations relative to the statically best design cannot be disputed, but is it absolutely necessary to take into account what we have learnt by hard experience. The track and the vehicle are not always in an irreproachable state of repair. Irregularities in the alignment of the track are the greatest enemies of sure guiding and they must be taken into account. We ought to be able to intercept the vibrations about the longitudinal axis of the locomotive. This condition can only be fulfilled by a spherical centre or longitudinal knife edge; here again these vibrations cannot be overcome except by auxiliary side bearing springs and by bearing springs on the bogie and in the main frame.

Considerations of a dynamic order result therefore in the conclusion that the bogie with lateral support of the frame on both sides is the best for guiding.

Opinions are divided as regards the dynamic behaviour of transverse equalisers at the leading end. They are not used on the guiding bogies of the steam locomotives included in this report ; on the other hand they are frequently found on the guiding bogies and pony trucks of electric locomotives. Jahn states (cf. *Zeitung des Vereins Deutscher Eisenbahnverwaltungen,* 1929, p. 1167) that « transverse equalisers on the leading wheels exercise an unfavourable action owing to the fact that at high speeds they limit the weight on the outside wheel on curves ».

In view of this state of affairs, it will be understood why the support of the main frame on the bogie by a centre pin — method of support Cm *— is represented in relatively few cases.*

By « total guiding pressure » (cf. the first sentence of this resumé) must be understood the guiding pressure due to the friction increased by the guiding pressure of the mass, as it is the total of these two pressures which forms the factor upon which safe running depends.

It is to be hoped that investigations into the guiding pressure of the mass may be made elsewhere so that the dynamic actions, partly unknown at present, of railway vehicles when running, may be better known in the future from the point of view of safety at high speeds.

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B. — THE TRACK.

Preamble.

If the problem of the relation between the vehicle and the track is to be solved so as to ensure safety at high speeds, the very nature of this investigation requires that the rolling stock and the permanent way should be considered separately from the aspect of the conditions which they should fulfil, in order that the running of the trains should be safe, fast, and also, after all is said and done, comfortable for the passenger. *The vehicle and the track are closely interrelated.* Stephenson, the pioneer of railways, had not failed to notice this important connection. He declared, after having recognised the part played by the track in the development of the power of the locomotive, that the rail and the wheel are tied together like « husband and wife ». If the « vehicle » and the « track » are treated here in two distinct parts, the considerations to which these relations give place will overflow at places from one part into the other. In the present part, concerning the track, we study the question as to how the permanent way is maintained and has to be maintained in view of the different factors summarised in the title, to give sufficient guarantees as to safety at high speeds.

First of all the meaning of the notion « high speed » had to be made precise in order to enable the administrations forming part of the International Railway Congress Association to reply to the questionnaire sent to them. Within the spirit of the investigation which we have here to conduct, a speed is considered high if

a) *on a straight alignment*

$V > 80$ km. (50 miles) per hour,

b) *on superelevated curves*

$V > 3.3 \sqrt{R}$ km. per hour,

in which R denotes the radius of the curve in metres, so that when

R =	200	250	300	400	500	600	700	800	900	1 000 metres,
V >	47	52	57	66	74	81	87	93	99	104 km. per hour.

[when

R =	10	12.5	15	20	25	30	35	40	40	50 chains,
V >	29.2	32.3	35.4	41	46	50.3	54	57.8	61.5	64.6 miles per hour.

] *on curves without superelevation* $V = 2.5 \sqrt{R}$ km. per hour, so that when

R =	160	180	200	225	250	300	400 m.
V >	32	34	35	38	40	43	50 km. per hour,

[

V >	19.9	21.1	21.7	23.6	25	30	40 m.
when R =	8	9	10	11 1/4	25	26.7	31 miles per hour,

]

when R =	500	600	700	800	900	1 000 metres,
V >	56	61	66	71	75	79 per hour.

[

when R =	25	30	35	40	45	50 chains,
V >	34.8	37.9	41	44.1	46.6	49 miles per hour.

]

In order to obtain high speeds, *a track of sufficient strength is required in which the rails are fastened to the sleepers in as uniform a manner as possible, laid upon ballast of a permanently elastic nature and a well drained, constantly dry formation.* Defects in the relative level of the rails, excessive sags at rail joints, variations in the gauge of the track over short distances, defective draining of the subsoil at bridge approaches are causes which, even in straight roads, constitute appreciable drawbacks to the smoothness and safety of running at high speeds, in spite of the use of vehicles suitable for these speeds. If the track is placed upon an elastic ballast and a dry subsoil, the swelling caused in winter by frost and which necessitates reduction in speed is not to be feared. On curves the safety is also compromised by jars at joints or by sudden changes in the radius of the curve, of the superelevation, of the gauge, and by superelevation ramps of irregular profile and excessive angle. Defects which exist in the superelevation inclines are particularly dangerous for the leading pair of wheels of a locomotive leaving a curve, because the removal of the load on the leading pair of wheels involves the risk of derailment. The majority of the railway administrations already pay special attention to the factors mentioned above : for superelevated curves, the variable quantity K of the equation giving the maximum permissible speed

$$V_{max} = k \sqrt{R}$$

increases generally as the radius and superelevation increase and approaches or attains the value K = 4 for radii R > 600, giving

$$\frac{V^2_{max}}{R} = 16.$$

As will be demonstrated further on by considerations on the superelevation, the relationship $\frac{V^2_{max}}{R}$ may also be expressed by the equation $\frac{V^2_{max}}{R} = K + 85\ h$, which takes into account the influence of the superelevation h in metres, while K represents a fixed quantity which varies between 5 and 8. On the average, we

have then : $\frac{V^2_{max}}{R} = 6.5 + 85\ h$. In curves without superelevation, those of turnouts for example, the speed is usually reduced so that the value

$$\frac{V^2_{max}}{R} = 8$$

is not exceeded except in isolated cases.

The enquiry has to be made as to *what is the speed* at which there is risk of *derailment on a curve*. A glance at the evolution of locomotive design within recent decades reveals the tendency to place the boiler higher and higher, which means that its centre of gravity is at the same time raised. Up to about 1890 the height of the centre line of express locomotive boilers did not as a rule exceed 2 m. (6 ft. 6 3/8 in.) above the rail, while in the case of modern boilers it is now about 3.40 m. (11 ft. 1 7/8 in.). As a result of this, the centre of gravity of the locomotive as a whole has been raised, in the course of years, from about 1.30 m. to 2.10 m. (4 ft. 3 13/64 in. to 6 ft. 10 9/32 in.) above rail level. It is obvious that for the same speed, curvature and superelevation, the degree of stability must decrease according as the centre of gravity of the whole of the locomotive is raised. *The risk of derailment of the guiding front pair of wheels towards the inside* of the curve is particularly great when, the authorised speed being appreviably exceeded, a locomotive passes from a circular curve on to a straight alignment immediately following it or from a circular curve of small radius on to one of considerably greater radius. The inner wheel with its load appreciably reduced, subjected to the vertical wheel pressure, then comes into contact, as a result of the rotational impulse, with the inner rail, with the horizontal guiding force Y. The ratio $\frac{Y}{Q}$ which is dealt with at length in the first part (Chapter IV) of this report, may have, under very unfavourable assumptions, a value varying between 1.00 and 1.25. Supposing a height of H = 2.00 m. of the centre of gravity of the complete locomotive above the top of the rail, a polar moment of inertia of the locomotive of $I_p = 150\,000$ kgr. sec^2; a pressure on the leading pair of wheels of G = 15 000 kgr. and a displacement of the centre of gravity of 0.11 m. for a superelevation h in metres and for a distance between the rolling circles of 1.50 m. the calculations gave :

for $\frac{Y}{Q} = 1.00$ the value $\frac{V^2}{R} = 24.3 + 85\ h$,

and

for $\frac{Y}{Q} = 1.25$ the value $\frac{V^2}{R} = 26.5 + 85\ h$.

The table which follows gives the values of $\frac{V^2}{R}$ corresponding to superelevations $h = 0$ to $h = 0.15$ m.

for $\frac{Y}{Q} = 1.000$ and $\frac{Y}{Q} = 1.25$.

$\frac{Y}{Q}$	$\frac{V^2}{R}$			
	h			
	0	0.05 m. (*1 31/32 in.*)	0.10 m. (*3 15/16 in.*)	0.15 m. (*5 29/32 in.*)
1.00	24.3	28.6	32.8	37.1
1.25	26.5	30.8	35.0	39.3

The above values for $\frac{V^2}{R}$ furnish a reference point for the lower limit of the speed at which a derailment may occur towards the inside when a circular curve is immediately followed by a straight alignment. They make it possible to appreciate *the importance of transition curves and of superelevation* as measures serving to increase security against derailment.

To ensure adequate security, another factor of great importance is the angle of the cant incline when a locomotive leaves a curve. On this subject we would refer to the very instructive studies of Dr.-Ing. Uebelacker (1). Since the passage of the locomotive body from one position of equilibrium to another sets up vibrations, it is possible that fairly great changes of load may occur if, as a consequence of the service or faults in construction and maintenance of the superelevation incline (or ramp) (Uberhöhungsrampe), the latter assumes a much greater angle than that to which it was initially adjusted. With a considerable unloading of the front directing wheel which is in contact with the outer rail, *derailment of the front axle towards the outside* is then to be feared. In order to ensure sufficient safety the German Railway Association has considered the necessity of curtailing the reduction of the inclination of the superelevation ramp according as the speed increases, by recommending in the « Technische Vereinbarungen » (Technical agreements), § 20, line 3, for the angle of the superelevation ramp the gradient $\frac{1}{n} = \frac{1}{10\,V}$; and, as a maximum, $\frac{1}{6\,V}$, in which V represents the maximum permissible speed in km./h. It is necessary however that the gradient should never exceed 1 in 300. In interpreting this instruction, it is found that, for the passage of a pair of wheels from the superelevated curve to the straight alignment, use is made for a superelevation of h metre, of the *same given time*, namely

in the case of $\frac{1}{n} = \frac{1}{10\,V}$, of the time $t = 36\ h$, and

in the case of $\frac{1}{n} = \frac{1}{6\,V}$, of the time $t = 21.6\ h$,

a time which, for example, for a maximum superelevation $h = 0.120$ m. (4 23/32 inches) is

with $\frac{1}{n} = \frac{1}{10\,V}$ $t = 4.32$ seconds, and

with $\frac{1}{n} = \frac{1}{6\,V}$ $t = 2.59$ seconds.

For the torsion of the locomotive as a whole, the total wheel base being a in metres, it is still necessary in theory to add the time $t = \frac{3.6a}{V}$ so that the total time becomes

with $\frac{1}{n} = \frac{1}{10\,V}$, $t = 36h + \frac{3.6a}{V}$, and

with $\frac{1}{n} = \frac{1}{6\,V}$, $t = 21.6h + \frac{3.6a}{V}$.

While in the past between wide limits of speed the same gradient was often given to the superelevation ramps, and in consequence the time available for the change of the equilibrium position of the locomotive as a whole decreased in proportion as the speed increased, the instructions quoted above have the object of making the *same* time available for the torsion of the locomotive on the superelevation ramp. The importance of the gradient of the superelevation ramp is tangibly demonstrated in these arrangements dictated by modern experience.

When a locomotive travels at a very high speed, without obeying the prescrib-

(1) « Die Anpassung der Lokomotiven und Tender an Gleisunebenheiten » (The adaptation of locomotives and tenders to inequalities in the track), *Organ für die Fortschritte des Eisenbahnwesens*. No. 20, 15 October 1928.

ed speed reductions, over a curve of relatively small radius, *it may overturn towards the outside* of the curve. According to calculations which have been made, supposing that the centre of gravity of the complete engine is situated at a height of 2 m. (6 ft. 6 3/4 in.) and that the displacement of the centre of gravity is 0.11 m. (4 11/32 in.) the locomotive will overturn when

$$\frac{V^2}{R} = 40.8 + 85\,h$$

For superelevations $h = 0$ to $h =$ 0.15 m. (5 29/32 inches), the following values are found for $\frac{V^2}{R}$

For $h =$	0	0.05 m. (1 31/32 in.)	0.10 m. (3 15/16 in.)	0.15 m. (29/32 in.).
we get $\frac{V^2}{R} =$	40.8	45.1	49.3	53.6

A typical case of the overturning of a locomotive towards the outside is described in *The Railway Engineer*, Vol. 52, No. 619, of August 1931. For the smallest radius of the actual curve R = 173 m. (8.65 chains) with an average superelevation of $h = 0.095$ m. (3 9/16 inches) we obtain, assuming a height of the centre of gravity of the locomotive H = 1.80 (6 feet) and a displacement of the centre of gravity $e = 0.09$ m. (3 1/2 inches) by means of a more exact equation, for the case of overturning :

$$\frac{V^2}{R} = \frac{95\,6 - 127\,e}{H} + 85\,h$$

wherein, after substituting the numerical values :

V = 97.4 km. (60.5 miles) per hour.

In the report of the accident the limits of speed between which the overturning must have occurred are estimated at 40 to 60 miles an hour (V = 64 to 97 km. per hour). The speed at which such a curve can be traversed without hesitation, allowing for the superelevation, is, according to the calculations made by the author, V = 50 km. or 31 miles an hour. This speed must have been greatly exceeded. The accident in question is most instructive and proves in addition that as a result of the big overturning moment, the existing check rails could not prevent the inside wheels from raising themselves above the check rail.

The following considerations moreover show that *check rails provide only a conditional means of safeguarding the working.* When a wheel is guided by a check rail, the point of pressure is brought further and further forward, if the diameter of the wheel is greater and and greater. The wheel makes contact with the check rail with a rough and not a smooth surface, i. e. with a high coefficient of friction, whereas in the absence of a check rail and in consequence in case of contact between the outside rail and the flange only, the point of pressure is only a very short distance from the centre line and in addition the guiding and sliding surfaces are polished and smooth, giving a low coefficient of friction. If the inside profile of the flange is perpendicular to the rim of the wheel and if an ordinary rail with the side of the head rounded off is used as a check rail, derailment is to be feared, even at relatively slow speeds, when the first pair of driving wheels of a 4-6-0 type locomotive with large diameter driving wheels does the guiding of the locomotive, either mainly or entirely at a comparatively great striking angle. As the guiding effort is applied in its greater part, no longer at the great distance from the first pair of wheels of the bogie to the centre of friction (cf. the first part of the report, Chapter I, Steam locomotives), but at

the lever arm, considerably shorter, which extends to the first coupled pair of wheels, a high guiding pressure is developed at the point of contact between the check rail and the inside face of the wheel which makes contact with it, i. e. at the point where the two convex surfaces, the rounded head of the rail and the rounded flange, touch one another, and the coupled axle climbs the side of the check rail. The *Bulletin of the Railway Congress reported,* in its issue of September 1931, p. 788, under the title of « Railway accident reports » two derailments which occurred, in spite of the use of check rails, on curves of comparatively small radii [R = 600 m. (30 chains) and R = 660 m. (33 chains)] having sufficient superelevation (h = 80 mm. (3 1/8 inches) and h = 76 mm. (3 inches). Although in the first case the speed had exceeded, with V=112 km. (70 miles) per hour the limit considered allowable of V = 89 km. (55 miles) an hour, no derailment ought to have occurred for this reason alone. In the second case, the speed at the moment of derailment was estimated at V = 80 to 88.5 km. (50 to 55 miles) an hour; it was then less than the speed at which the curve could have been traversed with adequate security and which, with V = 92 km. (57 miles) an hour could be regarded as perfectly admissible. These two accidents teach us that *the check rail can only offer a sufficient guarantee against derailment of a pair of wheels* especially in the case of the authorised speed being exceeded, when the check rail and the tyre have in relation to one another such a shape that the point of contact of the small diameter guiding wheel moves only slightly from its normal position, for as small as possible a striking angle and, that this may be so, the centring force acting at the leading bogie centre of the locomotive is made sufficiently great to prevent as far as possible the first coupled pair of wheels from exerting any guiding pressure on the check rail.

The above considerations concerning the relations between the speed, safety of traffic and the track show that in straight alignments the speed can be increased to any desired limit whatever, if the track is sufficiently strong, if it is well maintained in line and level, if the constituent parts are strongly interconnected, and if it rests on a uniformly elastic ballast bed and a well drained formation. In circular curves, it is specially necessary to maintain uniformity of curvature, of superelevation and of the gauge of the road and to provide for the joining of a straight and a curve by a superelevation ramp of uniform development and increasing gradient, on account of the risk of derailment towards the outside with increase of speed, and coinciding as far as possible with the transition curve. The interposition of a transition curve reduces appreciably the danger of derailment towards the inside, in the event of the maximum permissible speed being exceeded. The amount of superelevation allowed plays a very important part as regards the possible increase of speed on curves. The table below gives an appreciation of the limits of danger in the case of the maximum speed judged admissible for the standard gauge track being exceeded :

$$V = \sqrt{(6.5 + 85\,h)\,R}.$$

We have also expressed these limits as percentages, having assumed superelevations of $h = 0$ and $h = 0.120$ m. (4 23/32 inches) in order that the influence of the superelevation may be judged.

It will be seen from this table that, if we take as a basis a maximum allowable speed of

$$V = 2.55\sqrt{R}$$

for curves without superelevation, the lower limit of risk, in case the maxi-

Running speed V in km. per hour for a radius of curve R in metres and a superelevation h in metres of		Superelevation $h = 0$		Superelevation $h = 0.120$ m. (4 23/32 in.)		Superelevation $h = 0$		Superele $h = 0.1$ (4 23/3
		V km./h.	%	V km./h.	%	V km./h.	%	V km./h.
Maximum permissible speed.	$V = \sqrt{(6.5 + 85\,h)\,R}$	$2.55\sqrt{R}$	100	$4.09\sqrt{R}$	100	$2.55\sqrt{R}$	100	$4.09\sqrt{R}$
Risk of derailment to the inside through lack of a transition curve.	$V = \sqrt{(24.3 + 85\,h)\,R}$ $\left(\frac{Y}{Q} = 1.00\right)$	$4.93\sqrt{R}$	193	$5.87\sqrt{R}$	144	$4.93\sqrt{R}$	100	$5.87\sqrt{R}$
Risk of derailment to the inside through lack of a transition curve.	$V = \sqrt{(26.5 + 85\,h)\,R}$ $\left(\frac{Y}{Q} = 1.25\right)$	$5.15\sqrt{R}$	202	$6.09\sqrt{R}$	149	$5.15\sqrt{R}$	100	$6.09\sqrt{R}$
Risk of turning over to the outside.	$V = \sqrt{(40.8 + 85\,h)\,R}$	$6.39\sqrt{R}$	251	$7.14\sqrt{R}$	175	$6.39\sqrt{R}$	100	$7.14\sqrt{R}$

mum speed is increased at least 93 %, is relatively high, whereas with an assumed maximum permissible speed

$$V = \sqrt{(6\,5 + 85\,h)\,R}.$$

for superelevated curves the limit of risk is already reached with an excess of speed over the maximum of 44 % and, therefore, lies at a much lower value. On the other hand, the maximum may be increased with a superelevation $h = 0.12$ m. (4 23/32 inches) by 60 % relatively to the road without superelevation ($h = 0$), while in the region of the limits of risk the influence of the superelevation diminishes more and more.

In order to ascertain as reliably as possible how the track is actually maintained and is capable of better maintenance, from the various points of view examined in Chapters I to VII, so as to ensure a safety of operation appropriate to high speeds, we addressed a questionnaire to the member administrations of the International Railway Congress Association. We would express our best thanks to these administrations for their courteous replies. An examination of the replies has shown that the opinions on the principles involved are the same or nearly the same as regards a great number of issues, and that opinions seldom vary from the usual one on the subject of certain questions. The investigation which now follows is designed to provide, with the aid of the replies received and information gathered from other sources, a summary of the solutions to the questions of detail which have emerged from the principal questions as far as the track is concerned.

CHAPTER I.

I. — Strength of the permanent way.

From the point of view of the technical conditions of the track, the majority of the railway administrations classify the lines by taking as a basis the *axle load*.

$$= \frac{\text{tare} + \text{load}}{\text{no. of pairs of wheels of the vehicle.}}$$

On account of the adhesive weight required at starting, which is equal to the total weight on the coupled wheels, express steam locomotives of the 4-6-0 or 4-6-2 type, in use on the majority of the

systems concerned, have generally very heavy axle loads, varying between 17 and 20 tons. The

$$\textit{weight per linear metre} = \frac{\text{tare} + \text{load}}{\text{length of vehicle over buffers}}$$

is of importance, in conjunction with the axle loads, only for the calculation of bridges. For the classification of the lines from the point of view of the technical condition of the track the first things to be taken into consideration are *the maximum allowable speed and the frequency of the trains.* From the same point of view the German State Railways devide the tracks into :

First class lines, with intense traffic, high speeds and very heavy axle loads. In this group the most important lines carrying international traffic and rapid long-distance trains form a « special class ».

Second class lines, with average traffic and generally slower speeds, for all wheel loads.

Third class lines : all other lines.

A similar classification exists on the Netherlands Railways which place in the third group main lines carrying light traffic and secondary lines. Only in a few cases is attention paid to the gradients, the lay-out and other factors, suchs as for example, the classes of trains.

Two propositions have been presented for an international classification of lines. The Dutch East Indies State Railways make use of a « map of lines over which vehicles may be used » in which the permanent way and bridges are taken into consideration in such a way that the axle load and the load per unit of length are duly taken into account. By the aid of this, the allowable vehicle load may be easily and definitely determined. The German State Railways regard as practicable a classification of the lines according to *axle loads, speeds, number and weight of the trains* which use them.

The *calculation of the permanent way* is based, in principle, by the companies concerned, on two methods :

1. The calculation by Dr. *Winkler's* method with modifications of the coefficient (Couche's formula, North of Spain Railways) and substitution for the calculation of the weight of the rails in kgr. per metre (Sweden) and finally with amendments to bring the wheel base into account (Netherlands Railways according to the method recommended in 1930 by the German Railway Association in order to arrive at unifomity in the method of calculating the permanent way). In this method the effect of the ballast and of the formation is left aside, while allowance may be made for the speed.

2. The method of calculation due to Dr. *Zimmermann* on the one hand, with certain simplifications (Finnish State Railways, German State Railways), on the other hand with modification of the method according to the theory indicated by Mr. *Müller-Breslau,* in his work *Die graphische Statik der Bahnkonstruktionen,* Vol. II, 2nd part, 1925 edition, page 154, a theory which enables the effect of several pairs of wheels to be determined by means of influence lines, while Dr. *Zimmermann's* theory normally contemplates only a single pair of wheels because the calculation of the effect of several pairs of wheels is difficult and lengthy (Dutch East Indies State Railways). The influence of the ballast and of the formation is expressed in this method of calculation by the « seating coefficient » (ballast coefficient) while the effect of the speed en-

ters into the calcuation only in an indirect manner, so that the maximum allowable stress decreases in proportion as the speed increases.

The principal difference between these two methods of calculation lies then in the elimination or in the consideration of the deflection by the ballast coefficient. Since the permanent way engineers of the various administrations hold divergent opinions on the matter of the effect of the ballast coefficient on the calculated stresses, the more so since according as the value ballast coefficient increases, its influence on the stresses becomes more and more feeble, the German Railway Association and the German State Railways have undertaken trials with the object of finding the value of the ballast coefficient for different beds (ballast and formation) and of thereby discovering a means of calculation in which the calculated stresses agree in large measure with reality. These tests are at present in hand.

The fundamental formula for the calculation of the rail is

$$W = \frac{k_s \cdot G \cdot a \cdot k_v \cdot k_d}{\sigma_{max}}$$

in which

W represents the moment of the rail section in cm ^{3},

G = the wheel load, in kgr.,

a = the spacing of the sleepers, in cm.,

k_s = a coeficient for the static load,

k_v = a coefficient for expressing the speed,

k_d = a coefficient representing lateral forces or also dynamic forces,

σ_{max} = the greatest allowable stress in kgr. per cm^2.

The coefficient k_s is, according to *Winkler*, $k_s = 0.1888$, according to Couche (North of Spain Railways), 0.192. As a basis of comparison, for a beam on two supports loaded at the centre with a concentrated load, it would be found that $k = 0.25$. According to the *wheel base formula* recommended by the German Railway Association we obtain for the intermediate pairs of wheels

$$k_s = \frac{12m \cdot n - 7(m+n) + 4}{16[3m \cdot n - (m+n)]}$$

and for the end pair of wheels

$$k_s = \frac{12n - 7}{16(3n - 1)}$$

denoting by m and n the ratio of the wheel base on each side of the wheel load, to the sleeper spacing. The value k varies generally between

$k_s = 0.21$ for short wheel bases and

$k_s = 0.24$ for long wheel bases.

In *Zimmermann's* calculations, the development of which we follow here as it appears in his work « Die Berechnung des Oberbaues » (Calculations for the permanent way),

$$k_s = \frac{8\gamma + 7}{16\gamma + 40},$$

and thus depends upon the ratio

$$\gamma = \frac{B}{D} = \frac{\text{Degree of stiffness of the rail}}{\text{Degree of rigidity of the supports}}.$$

Assuming the same section of rail and the same spacing of sleepers γ becomes correspondingly smaller as the bed is more rigid and as, consequently, the ballast coefficient C and the length and width of the sleepers become greater. We have for

γ =	0.2	0.6	1.0	2.0	3.0	4.0
k =	0.199	0.238	0.268	0.319	0.352	0.375

The values k_s obtained by *Zimmermann's* method are only exact for a wheel load which is placed in the middle of a span with four elastically flexible supports, equally spaced, i. e. for an

isolated wheel load. Now for a short wheel base, there come into play the several wheel loads, so that the bending moments are reduced. According to the researches carried out on the latter case by Loewe, it appears that

for γ =	0.2	0.6	1.0	2.0	3.0	4.0
for an intermediate pair of wheels k_s =	0.187	0.183	0.182	0.185	0.193	0.204
for an end pair of wheels k_s =	0.193	0.213	0.227	0.253	0.271	0.286

From a comparison of the coefficients k_s, it may be concluded that *Zimmermann's* calculation is applicable only to individual free pair of wheels, i.e. in the first instance to four-wheeled vehicles with a long wheel base. But it is also applicable to an individual locomotive trailing carrying pair of wheels situated at a comparatively great distance between the last coupled pair of wheels and the leading pair of tender wheels. For the outer pair of wheels of locomotives, tenders and bogies of short wheel base and for the intermediate pairs of wheels, especially the intermediate pairs of locomotives and tenders, calculations founded on the « wheel base formula » of the German Railway Association and according to Loewe give more acceptable results which, for intermediate pairs of wheels with short wheel base, approach the coefficient of Winkler.

We will note therefore, in the analyses which follow, that the coefficient k_s depends essentially on the *spacing of the axles* and increases as the spacing of the axles increases. If the latter is small, it approaches the value k_s = 0.188; if it is great, it approaches the values k_s = 0.375 (equal approximately to 2× 0.188). As average values we may write :

for individual pairs of wheels : k_s = approx. 0.290;
for outer pairs of wheels : k = approx. 0.240;
for intermediate pairs of wheels: k = approx. 0.190.

The coefficient k_v, which expresses the effect of the speed, represents the ratio between the greatest allowable stress σ_{max} in kgr. per cm² and the permissible stress for a given speed σ_v in kgr. per cm². We may then write :

$$k_v = \frac{\sigma_{max}}{\sigma_v}.$$

Winkler was the first to make allowance for the speed in calculating the bending moment on the rail in terms of the speed

$$M_v = M_s \left(\frac{1}{1 - M_s \frac{v^2}{EIg}} \right) = M_s k_v$$

in which

M_v = 0.1888 G_a the bending moment for a static load,
v in cm. per sec. = 28 V in km./h., the speed,
g the acceleration due to gravity = 981 cm. per sec²,
E the modulus of elasticity in kgr. per cm²,
I the moment of inertia of the rail in cm⁴.

On substituting the numerical values, the equation

$$k_v = \frac{1}{1 - M_s \frac{v^2}{EIg}}$$

becomes, for light rail sections [33 kgr. per metre (66.5 lb. per yard), I = approximately 1 000 cm⁴ (24.026 in⁴)] with a fairly wide sleeper spacing [70 cm. (2 ft. 3 1/2 in.)]:

$$k_v = \frac{1}{1 - \frac{V^2}{30\,000}} = \frac{30\,000}{30\,000 - V^2}$$

and for heavy sections [45 kgr. per metre (90.7 lb. per yard) I = approximately 1 600 cm⁴ (38.44 in⁴)] with smaller sleeper spacing [60 cm. (1 ft. 11 5/8 in.)]:

$$k_v = \frac{1}{1 - \frac{V^2}{40\,000}} = \frac{40\,000}{40\,000 - V^2}.$$

These equations show that in the first case, for V = 173 km. (107.5 miles) per hour and in the second for V = 200 km. (124 miles) per hour, the value of $k_v = \infty$, that consequently their range of application is limited, but that in any case they have sufficed for the range of speeds formerly common. Moreover in his work published in 1875, Winkler was only concerned with speeds of 9 German miles, say (42 statute miles) per hour. The Finnish State Railways take account of the speed according to Winkler's method while they calculate the static wheel load by Zimmermann's method.

The Dutch East Indies State Railways express the speed by a coefficient having a mean value of

$$k_v = 1 + \frac{V}{200}$$

The Swedish Railways take

$$\sigma_v = 1100 - 5\,V,$$

giving

$$k_v = \frac{1\,100}{1\,100 - 5\,V} = \frac{1}{1 - \frac{V}{220}}.$$

On the German State Railways

$$\sigma_v = 1\,300 - 2\,V,$$

$$\text{giving } k_v = \frac{1\,300}{1\,300 - 2\,V} = \frac{1}{1 - \frac{V}{650}}$$

The Netherlands Railways use the coefficient

$$k_v = 1 + \frac{V^2}{30\,000},$$

obtained as a result of very numerous observations made by the Netherland Railways and the Dresden Direction of the German State Railways and recommended by the German Railway Association.

It goes without saying that by reason of the diversity of form of the equations for k_v the numerical values corresponding to an increasing speed vary between wide limits. The table which follows, of the numerical values for k_v at different speeds, illustrates these variations.

$$k_v = \frac{\sigma_{max}}{\sigma_v}$$

V km. (*miles* per hour).	$\frac{1}{1 - \frac{V^2}{30000}}$	$\frac{1}{1 - \frac{V^2}{40000}}$	$1 + \frac{V^2}{30000}$	$1 + \frac{V}{200}$	$\frac{1}{1 - \frac{V}{220}}$	$\frac{1}{1 - \frac{V}{650}}$
40 (25)	1.06	1.04	1.05	1.20	1.22	1.07
60 (37.3)	1.14	1.10	1.12	1.30	1.38	1.10
80 (50)	1.27	1.19	1.21	1.40	1.57	1.14
100 (62)	1.50	1.33	1.33	1.50	1.83	1.18
120 (75)	1.92	1.56	1.48	1.60	2.20	1.23
150 (93.2)	4.00	2.29	1.75	1.75	3.14	1.30

If we neglect the first column of the values of k_v, which are applicable only to light section rails, the limiting values are

for V = 100 km. (*62 miles*) per hour, k_v = 1.18 to 1.83
for V = 120 km. (*75 miles*) per hour, k_v = 1.23 to 2.20
for V = 150 km. (*93.2 miles*) per hour, k_v = 1.30 to 3.14.

The coefficient k_d *is* used by the Dutch East Indies State Railways to allow for horizontal forces ; it is

for the locomotives, k_d = 1.30
for the carriages, k_d = 1.15

The Madrid-Saragossa-Alicante Railways make allowances for the dynamic forces by inserting, according to Ast's method, the dynamic wheel loads with G = 2.4 whence

$$k_d = 2.4$$

The *greatest allowable stress* σ_{max} in the rails varies between exceptionally wide limits, namely between σ_{max} = 700 kgr. per cm² (9 956 lb. per in²) on the North of Spain Railways, as the lower limit, and σ_{max} = 3 500 kgr. per cm² (49 780 lb. per in²), making allowance for dynamic actions, on the Madrid-Saragossa-Alicante Railways, as the upper limit. The decisive effect of the assumption regarding σ_{max} for the computation of W is therefore evident. Several administrations allow, for worn rails, on secondary lines, stresses greater by 10 to 20 % than for those allowed on new rails. The mean values of the greatest allowable stress are generally comprised between σ_{max} = 1 100 to 1 500 kgr. per cm² (15 645 to 21 335 lb. per in²); the average value is therefore about σ_{max} = 1 300 kgr. per cm² (18 490 lb per in²). Very high stresses are tolerated only when it is necessary to use high coefficients k_v and k_d or k_v-k_d. [Dutch East Indies State Railways, σ_{max} = 2 075 kgr. per cm² (34 510 lb. per in²), Finnish State Railways, for new rails σ_{max}=2 000 kgr. per cm² (28 445 lb. per in²), Madrid-Saragossa-Alicante Railways σ_{max} = 3 500 kgr. per cm² (49 780 lb. per in²)]. For steel sleepers, a single railway administration, the German State Railways, has communicated the greatest allowable stresses. It is σ_{max} = 1 400 kgr. per cm² (19 910 lb. par in²) on lines for through and express trains ; σ_{max} = 1 500 kgr. per cm² (21 335 lb. per in²) on lines without through or express trains. The Netherlands Railways allow a load of 12 000 kgr. (26 450 lb.) per sleeper in the light of the calculation for the permanent way recommended by the German Railway Association. No special values have been given for the switch tongues.

The coefficients k_s, k_v *and* k_d in the numerator and the *greatest allowable stress* σ_{max} in the denominator of the equation

$$W = \frac{k_s \cdot G \cdot a \cdot k_v \cdot k_d}{\sigma_{max}}$$

naturally exert much influence on the final result and, in consequence, on the numerical value of the moment of resistance which has to be calculated. The determination as exactly as possible, of the coefficient k_s often necessitates long and careful work with which the assumptions relative to k_v, k_d and σ_{max} do not agree, assumptions generally speculative and in the long run, often arbitrary and approximate, which definitely influence the final result. If the original premises are that between the moment of resistance W in cm³ and the weight of the rail g in kgr. per metre, there is a ratio applicable also very closely to worn rails the moment of resistance and weight of which are less, a ratio which may on the average be expressed by

$$\frac{W}{g} = \text{approx. } 4.5$$

the equation relative to the moment of

resistance becomes that relative to the weight of the rail

$$g = \frac{k_s \cdot k_v \cdot k_d}{4.5 \cdot \sigma_{max}} \; Ga$$

whence, if we put

$$\frac{k_s \cdot k_v \cdot k_d}{4.5 \cdot \sigma_{max}} = \frac{1}{K}$$

$$g = \frac{G \cdot a}{K}.$$

The numerical value of the quantity

$$K = \frac{Ga}{g} = \frac{4.5 \cdot \sigma_{max}}{k_s \cdot k_v \cdot k_d}$$

is given in the table below where G represents in kgr. the heaviest load per wheel of the locomotive, which is to be found in practice.

No.	Administration.	G kgr. (*lb.*)	*a* cm. (*ft. in.*)	Ga kgr. cm.	*g* kgr./m. (*lb. per yard*)	K = G/g
1	Danish State Rys	8500 (*18740*)	63 (*2' 3/4"*) 66 (*2' 2"*)	535 500 561 000	37 (*74.6*) 45 (*90.7*)	14 473 12 467
2	German State Rys.	10000 (*22040*)	65 (*2' 1 5/8"*)	650 000	49 (*98 8*)	13 265
3	Finnish State Rys.	6300 (*13890*)	72 (*2' 4 3/8"*)	453 600	43.6 (*87.9*)	10 404
4	Netherlands Rys.	9200 (*20280*)	60 (*1' 11 5/8"*)	552 000	46 (*92.7*)	12 000
5	Dutch Indies State Rys.	6400 (*14110*)	69 (*2' 3 1/8"*)	441 600	33.4 (*67.3*) 41.5 (*83.7*)	13 222 10 641
6	Norwegian State Rys.	7150 (15760)	66 (*2' 2"*)	571 900	49 (*98.8*)	11 060
7	Portuguese Ry. Co.	8000 (17640)	61 (*2' —*)	488 000	45 (*90.7*)	10 844
8	Swedish State Rys.	8500 (18740)	66 (*2' 2"*)	561 000	43.2 (*87.1*)	12 986
9	Grängesberg-Oxelösund (Sweden)	8000 (17640)	77 (*2' 6 5/16"*)	616 000	41 (*82.7*)	14 293
10	Bergslagen Ry.	8900 (19620)	69 (*2' 3 1/8"*)	614 000	45.3 (*91.3*)	13 556
11	Swiss Federal Rys.	10200 (22480)	60 (*1' 11 5/8"*)	612 000	46 (*92.7*)	13 304
12	North of Spain Rys.	8500 (18740)	63 (*2" 3 4"*) 75 (*2' 5 1/2"*)	535 500 637 500	42.5 (*85.7*)	12 600 15 000
13	Madrid-Saragossa-Alicante	8100 (17860)	63 (*2' 3 1"*)	510 300	45 (*90.7*)	11 340

The value of K is seen to vary between K = approx. 10 000 and K = approx. 15 000, the mean value is approx. K = 12 500. We will now extend the comparison by including the latest particularly heavy American rails used for some time by the Pennsylvania Railroad and the Kansas City Southern Railway. One of these rail sections, weighing 64 kgr. per metre (130 lb. per yard) is intended for a wheel load of 18 000 kgr. (39 700 lb.), for a speed of 122 km. (76 miles) per hour, and the other, weighing 75 kgr. per metre (152 lb. per yard) for wheel loads of 22 500 kgr. (49 500 lb.) with a speed of 152 km. (94.5 miles) per hour. The values of K are shown in the following table.

Administration.	G kgr. (*lb.*)	*a* cm. (*ft. in.*)	Ga kgr. cm.	*g* kgr/m. (*lb. per yard*)	$K = \frac{Ga}{g}$
Pennsylvania Rd. and Kansas City Southern Ry.	18 000 (*39 700*)	51 (*1' 8 1/16"*)	918 000	64 (*130*)	14 344
	22 500 (*49 500*)	51 (*1' 8 1/16"*)	1 147 500	75 (*152*)	15 327

From the preceding tables, the conclusion may be drawn that with a reasonable amount of wear of the rails, *the weight of the rail for high-speed lines* is comprised between the limits

$$g = \frac{Ga}{13\,000} \quad \text{to} \quad \frac{Ga}{15\,000}$$

The expression $K = \frac{Ga}{g} = \frac{4.5 \cdot \sigma_{max}}{k_s\, k_v\, k_d}$ shows further that in the event of worn rails being re-used, by allowing a higher stress in lines traversed at a lower speed, that is to say with a reduction of k_v, the value of K increases. For reasons of an economic nature, we might then allow

$$g = \frac{Ga}{15\,000} \quad \text{to} \quad \frac{Ga}{17\,000}$$

to enable the further utilisation of old light sections of low weight *g* for heavy loads G by reducing the sleeper spacing. Experiments directed towards the exact measurement of the effective stress in the rail, taking account of the ballast and the formation, of the arrangement of the axles and the speed, are in progress; we may therefore expect enlightment on the values of k_s, k_d and σ_{max} and be able to calculate the value of K more exactly.

Up to the present *the effect of the longitudinal forces acting upon long rails* is not taken into consideration when calculating the permanent way. By reason of the usually low maximum stresses upon which the calculation of the rails is based, the material of the rails can be taken as not being stressed to excess by the longitudinal efforts which may be set up, provided that in the type of permanent way used the rails are fastened to the sleepers as rigidly as possible, and that the ballast offers sufficient resistance to any movement of the sleepers upon it.

For heavy wheel loads (G = 8 to 10 tons) and high speeds [V = 80 to 120 km. (50 to 75 miles) per hour], *rails having a weight* $g = 40$ to 45 kgr. per metre (80.6 to 90.7 lb. per yard) are generally used, and also in recent times $g = 49$ kgr. per metre (98.8 lb. per yard). The weight of the rails in use by the administrations concerned are shown in the table giving the values of K (on page 2350). For a given wheel load G in kgr. and a sleeper spacing averaging $a = 65$ cm. (2 ft. 1 5/8 in.) we may determine approximately the necessary weight of rail by means of the equation

$$g = \frac{Ga}{K} = \frac{G \cdot 65}{13\,000}$$

We then get :

for G =	8 000	8 500	9 000	9 500	10 000	kgr.
g =	40.0	42.5	45.0	47.5	50.0	kgr. per m.
g =	80.6	85.7	90.7	20 940	22 040	lb.
for G =	17 640	18 740	19 840	95.7	100.8	lb. per yard

i. e. rail weights which differ but little from those in actual use.

The length of the rails is usually from 12 to 18 m. (39 ft. 4 1/2 in. to 59 ft. 5/8 in.). *Long rolled rails* 24 m. (78 ft. 9 in.) in length are employed by the Swiss Federal Railways in tunnels and the German State Railways have laid some of 30 m. (98 ft. 5 1/8 in.) length on the lines carrying international traffic and rapid and long-distance trains. In addition, *long welded rails* [28.80 m. (94 ft. 6 in.) or 30 m. (98 ft. 5 1/8 in.)], formed from two rails of ordinary length are in use on the Danish State Railways, the Norwegian State Railways and the Bergslagen Railway (Sweden). Some welded rails 60 to 90 m. (196 ft. 10 in. to 295 ft. 3 in.) long formed of four to six rails of ordinary length have been laid as a trial on the Norwegian State Railways. The German State Railways have recently placed in the road in order to test them under traffic conditions, some rails 60 m. *long* formed by welding together two rolled rails 30 m. long. The great advantage of long rails consists in the *elimination of joints* which, being the weak point of the road, detract on the one hand from smooth running, and also in certain cases, with a faulty setting and a high speed, from safety of operation, and on the other hand, exercise a prejudical action by causing wear of the track and rolling stock.

The *sleeper spacing* is, for intermediate sleepers from 77 to 60 cm. (2 ft. 6 5/16 in. to 1 ft. 11 5/8 in.).

Spacings exceeding 70 cm. (2 ft. 3 1/2 in.) are met with only in isolated cases while a general tendency is noticed to reduce the sleeper spacing to about 65 to 60 cm. (2 ft. 1 5/8 in. to 1 ft. 11 5/8 in.). The sleepers are generally placed closer together at the joint than under the body of the rail; with an overhanging joint their spacing is from 42 to 58 cm. (1 ft. 4 1/2 in. to 1 ft. 10 13/16 in.). More than half the administrations which have replied to this questionnaire have adopted *the joint with double sleeper* obtained by closing in the joint sleepers, which however does not affect the elasticity of the short rail ends projecting beyond the sole plates. The Netherland Railways use, in addition to the overhanging joint, an unusual type of joint with a *single sleeper*, in which the *rail ends are held up in a chair* by the wedging action of fish plates of appropriate shape, in such a way that there is a distance of 20 mm. (25/32 inch) between the bottom face of the rail and the upper face of the bed of the chair.

On account of the amount of timber available in the majority of countries wood *sleepers* are principally used; the Dutch East Indies State Railways and the German State Railways use steel sleepers in addition to timber ones. The Swiss Federal Railways in general use only metal sleepers; it is only in the tunnels that they substitute timber ones. The *rails are fastened down* on most of the railways which have answered this questionnaire, in the familiar manner, i. e. on timber sleepers by the use of flat sole plates, sometimes also by plates with a tenon, and coachscrews, on metal sleepers by holding down plates and hook bolts; the Dutch East Indies State Railways use in addition sole plates with a tenon. Five railways have adopted the practice of *separate fixing of the rail* to the sole plate and the plate to the sleeper. The Netherlands Railways and the Swiss Federal Railways use with this object cast iron chairs screwed to timber sleepers and on which the rails are fixed separately by means of clips and hook bolts. The German State Railways, the Norwegian State Railways and the Bergslagen Railway, Göteborg (Sweden) have adopted for use with timber sleepers rolled ribbed sole plates which are fixed to the sleeper by means of four coachscrews, while the separate fastening of the rail to the ribbed sole

plate is assured by locking plates and hook bolts, the hooked ends of which engage in slots cut in the ribs of the plates. The principle of adopting *one standard form of fastening for both timber and metal sleepers* has led the German State Railways to weld the ribbed sole plates to the metal sleepers. This procedure possesses a dual advantage : firstly it avoids drilling the sleepers with its undesirable consequences, and secondly the sleeper is reinforced at the rail supports where the stresses are greatest.

The *ballast* which primarily has to transform the energy of the shock due to the rolling loads into work of elastic deformation, and which may thus be regarded in the light of a spring interposed between the permanent way and the formation ought, if it is to perform this duty efficiently, to be formed of suitable material and have an adequate thickness. Although it has been generally recognised that broken stone which weathers well and resists shocks constitutes the best form of ballast, the majority of the countries on account of economic considerations find themselves under the necessity of using principally those materials for ballast which are available locally. Likewise, *the utility of as thick a layer of ballast as possible* is recognised as the result of practical experience and theoretical considerations which indicate that the depth of the ballast, measured from the upper face of the sleepers, should be equal to the distance from centre to centre of the sleepers. But here again financial considerations play a decisive role, with the result that this desirable condition is only approximately fulfilled on a small number of systems. In consequence, the depth of the ballast varies on the railways which have replied to the questionnaire from 33 to 62 cm. (13 to 24 3/8 in.), its mean value is about 45 cm. (17 11/16 inches).

When the newly placed ballast is tamped, a procedure which is still adopted to a preponderating extent, the necessary consolidation of the ballast is brought about by the rolling loads, the main result of which is an unstable road, without counting the other bad features which result therefrom. These objections may be counteracted by *ramming or rolling the ballast.* The ballast is thus artificially reduced to a condition which up to the present it has been left to the traffic to produce. For the ballast considered as a spring, the result is a raising of the elastic constant or of the « ballast coefficient ». Ramming is employed on the Netherlands Railways and on the Dutch East Indies State Railways, rolling on the Swiss Federal Railways and on the Danish State Railways; both methods are used by the German State Railways.

A feature of preponderating importance in the attainment of high speeds with every guarantee of safety is a *formation* which is well drained and maintained in a constantly dry condition. A road of great strength, well braced and resting upon a thick bed of stone ballast, of perfect permeability, will nevertheless be always unstable and sometimes necessitate, even in straight alignments, a reduction in speed on account of the irregularities which it will develop if the formation under the action of moisture has lost its resistance and its elasticity and has become plastic. The subsoil layers work their way into the ballast and make it dirty, so that it becomes inelastic; moreover in winter they facilitate the swelling of the formation due to frost. Consequently the railway administrations are increasingly giving special attention to the *drainage and drying up* of the formation. If the side ditches do not lead the water away and stagnant water is found to accumulate, adequate drainage must be provided. The lowering of the water level to 1.50 m. (5 feet) at least below the top of the rails by excavating the

bottom of the ditches and placing in position drain pipes filled with coke will ensure a permanently dry formation. Transverse cross drains of the same kind, filled flush with sand or coke have given equally good results in eliminating wet places in the road. As a general rule, the administrations who answered the questionnaire improve unfavourable conditions of the subsoil by placing layers of sand or coke under the ballast, after having provided an adequate drainage system. The insertion of wooden packings between the sole plates and the sleepers when the road has been broken up by frost, a method adopted on the Swedish Railways, constitutes an expedient which remedies the effect but not the cause. In Sweden also the lowering of the water level and the use of drain pipes, the introduction of good ballast, and the replacement of the excavated wet subsoil by coke are considered to be the best means of preventing the road being disturbed by frost.

The results of the preceding considerations may be summarised as follows :

The rails are generally calculated by different methods based on the heaviest wheel load and the sleeper spacing, and taking into account the speed. The weight of the rails in kgr. per metre, the ratio of which to the moment of resistance of the transverse section of the rail varies but little, depends in the first instance, despite the diversity of the methods of calculation used, upon the wheel load and the sleeper spacing, as has been demonstrated by comparative computations. Long rails fastened as rigidly as possible to sleepers resting upon stone ballast reduce the number of joints. The reduction of the sleeper spacing, especially at the joints, by the use of the double sleeper joint, the independent fastening of the rail to the sole plate and of the latter to the sleeper, a rammed or rolled ballast, as thick as possible, on a well drained formation, permanently in a dry state, are suitable means by which to reduce the deflection and the sinking of the joint of the two running rails to such a degree that, at high speeds, variations in pressure of the leading wheels can occur only between narrow limits.

CHAPTER II.

Gauge widening on curves.

The standard track gauge on straight alignment has been the result essentially of the growth of railways. This is why most administrations adopted the figure of 1.435 m. (4 ft. 8 1/2 in.) originally used in England). For economic reasons, a radical modification of the gauge to increase safety at high speeds could not be entertained. Gauges other than the standard have been adopted by the following administrations :

Dutch East Indies State Railways .	1.067 m. (3 ft. 6 in.)
Finnish State Railways	1.524 m. (5 feet)
Portuguese Railways	1.665 m. (5 ft. 5 35/64 in.)
North of Spain Railways	1.674 m. (5 ft. 6 in.)
Madrid-Saragossa-Alicante Railways.	1.674 m. (5 ft. 6 in.)

The standard gauge is generally *measured* at 12 to 15 mm. (15/32 to 19/32 inch) [most often 14 mm. (35/64 inch)] below the rolling surface of the rail.

The *tolerances* vary from —3 mm. to +10 mm. (—15/128 to +25/64 inch) sometimes +15 mm. (+19/32 inch).

The *play between the pair of wheels and the track* on the straight is essential for safety and smooth riding at high speeds. On the straight the pairs of wheels advance along an approximately sinusoidal path. At high speeds, this motion may result in uncertain and unstable running if the play between the wheels and the track is to great and

as a result the striking angle increases. On the other hand when too little play is allowed, the tyre and the rail constantly rubbing together under pressure, irritating noises are caused and may even result in derailment. *The theoretical investigations and practical experiments of recent years tend to show that a reduction of the play habitually provided up to the present between the wheels and the rail on the straight may have advantages and certainly so from the point of view of smooth riding, at the same time as it increases safety by reducing the cutting angle.* The smallest amount of play reported by the administrations consulted is 4.5 mm. (83/128 inch) and the greatest 15 mm. (19/32 inch).

The *thickness of the flange,* measured at 10 mm. (25/64 inch) outside the rolling circle, is 20 mm. (25/32 inch) minimum and 34 mm. (1 11/32 inches) maximum.

The *height of the flange* beyond the tread circle is on the majority of the systems 25 mm. (63/64 inch) minimum and 36 mm. (1 27/64 inches) maximum.

The *gauge widening on curves* is generally fixed according to practical experience and to give sufficient play for the wheels of the most unfavourable vehicle. The Grängesberg-Oxelösund Railways (Sweden) puts forward a formula which can however be considered as approximate only, being based on purely geometrical considerations.

In comparing the numerical values communicated by the administrations who replied to the questionnaire and which were calculated as a function of the radius of the curve, it is recognised that the systems may in fact be divided into two groups, one of which does not prescribe any gauge widening for radii of curves $R > 800$ to 1 000 m. (40 to 50 chains), and the other for radii $R > 300$ to 500 m. (15 to 25 chains). The values of the first group correspond on the average to the formula:

$$e_{mm} = \frac{(1\,000 - R)^2}{30\,000}$$

while the values of the second group are, on the average, lower by 10 mm. (25/64 inch) than those of the first group.

From the point of view of safety of working at high speeds, care must be taken to widen the track on curves as little as possible as in this way the resulting cutting angle between the wheel and the rails is a minimum.

As relative *tolerances on the gauge widening laid down,* the bulk of the administrations authorise a reduction of 2 to 3 mm. (5/64 to 15/128 in.) and an increase of 5 to 10 mm. (25/128 to 25/64 inch), going in one case up to 15 mm. (19/32 inch).

Everything seems to show that the *increasing gauge widening* (on passing from the straight to the curve) should be made to depend upon the speed. This point of view has only been adopted by the Portuguese Railways who lay down for high speeds an increase of 0.5 mm. per metre only (1 in 2000), but for lower speeds 1 mm. per metre (1 in 1000) and at turnouts 4 mm. per metre (1 in 250).

The other administrations only lay down *a single value* for the gauge widening, as allowance has already been made indirectly for the speed by the radius of the curve. The simple stepped rate of increase (from 2 to 5 mm.) suffices if the beginning and the finish of the transition are settled; in the contrary case, a uniform growth is imposed of 1 mm. to 3 mm. at the most, per metre.

The *gauge widening at turnouts* varies very considerably, even allowing for the radii of the curves. This fact indicates that, in the construction of turnouts, the bases for deciding the gauge widening are generally more controversial than

in the construction of the track itself. The following figures have been given:

at the point of the tongue : most commonly 5 to 10 mm. (25/128 to 25/64 inch), more seldom 0 to 15 mm. (0 to 19/32 inch);

on the straight at the heel of the tongue : most frequently 0 mm.;

in the lead at the heel of the tongue : most commonly 0 to 6 mm. (15/64 inch), more seldom 12 to 15 mm. (15/32 to 19/32 inch);

in the lead between the heel of the tongue and the nose of the crossing: most commonly 6 to 15 mm. (15/64 to 19/32 inch), more rarely 0 mm. and up to 25 mm. (63/64 inch);

for crossings on curves, at the nose of the crossing: as in the lead between the heel of the tongue and the crossing.

Numerous theoretical investigations and practical experiments on the reduction of the gauge widening through curves, accounts of which appear in the technical press, have only resulted up to the present in slight modifications of the prescribed rules. The information bearing upon the experiments on gauge widening, sometimes to a very small extent, indicates however that endeavours are being made to ascertain the effects of a reduction in the gauge widening. Economic questions take the first place when calculating the amount of gauge widening.

CHAPTER III.

Radius of curves.

The *maximum authorised speeds* for different radii of curves are generally comprised between $V = 3\sqrt{R}$ and $V = 4\sqrt{R}$. The invariable result of this is, independently of the superelevation, that there are greater variations in speed on the curves of greater radii. Thus, the curve of radius R = 450 m. (22 1/2 chains) is traversed on one system at V = 60 km. (37.3 miles) per hour, and on another at V = 85 km. (52.8 miles) per hour, or the curve of R = 600 m. (30 chains) with V = 65 km. (40.4 inches) and V = 100 km. (62 miles) per hour.

On the other hand, *the maximum speeds authorised for varying gradients* do not differ very appreciably on the greater number of the systems. A remarkable fact is that the lowest values for the maximum allowable speed on down gradients are reported by one administration (Netherlands Railways) whose lines are located principally in flat country, and the greatest values by one administration (Swiss Federal Railways) the greater part of whose lines are mountain lines.

These maximum speeds are usually fixed as a result of practical experience; in a few instances only have theoretical considerations been taken into account. *In each case, when calculating the maximum permissible speeds, there are definite relationships between the highest authorised speed* V_{max} *in* km./h., *the radius of the curves* R *in metres, and the superelevation* h *in metres, relationships which are given by the fundamental law*

$$h = u \cdot \frac{V^2_{max}}{R}$$

from which we get $V_{max} = \sqrt{\frac{h\,R}{u}}$

in which u *is a coefficient.* From the moment, then, that the value of the superelevation as a function of the radius of the curve is determined, a value which depends in its turn on the gradient of the superelevation ramp and on the length of the transition curve coinciding with the superelevation ramp, and if, in addition, the coeficient *u* is known, the speed may be calculated by means of the preceding formula.

Putting for example, $u = 0.008$, we have :

for $h =$	0.100 m.	0.120 m.	0.130 m.
$V_{max} =$	$3.54\sqrt{R}$	$3.87\sqrt{R}$	$4.03\sqrt{R}$.

The formula presupposes the introduction of a finite value for the superelevation h. In practice, in order to reduce the numerical factor which precedes the radical, a useful reduction in view of the shortening of the transition curve when the superelevation diminishes for a smaller curve radius, we have taken

$$V_{max} = 4\sqrt{R - 50}$$

an expression in which, however, the radii of curves $R < 50$ m. are not taken into consideration from the point of view of the speed. A more recent formula obtained by calculation.

$$V_{max} = \frac{4\,R}{\sqrt{R + 50}}$$

makes it possible to determine the speed on standard gauge lines for small radii of curves. But in the two formulæ, the value of the superelevation h does not appear ; it is taken account of only indirectly. In the considerations set forth in the chapter « Superelevation » to which we would refer here, the *influence of the superelevation on the maximum allowable speed* is considered. From the point of view of smooth riding we have established, for the superelevation h in millimetres the formula :

$$h = 11.8\,\frac{V^2}{R} - 61$$

from which is derived

$$V_{max} = \sqrt{\frac{(h + 61)\,R}{11.8}}$$

From this we get

for $h =$	100 mm.	110 mm.	120 mm.
$V_{max} =$	$3.69\sqrt{R}$	$3.81\sqrt{R}$	$3.92\sqrt{R}$,

i. e. slightly higher values. *This formula demonstrates the utility of an ample superelevation to obtain greater smoothness of running and an increase of safety on curves at high speeds.* From the point of view of adequate stability by using the formula

$$h = 11.8\,\frac{V^2_{max}}{R} - 90$$

quoted in the chapter « Superelevation » the *maximum permissible speed on non-superelevated curves*

$$V_{max} = 2.76\sqrt{R}$$

can also be calculated.

The smallest radius of curve on lines run over at high speed is almost always fixed, on running lines and sometimes also through turnouts, at $R = 300$ m. (15 chains). Everything possible is done to increase this to $R = 500$ up to 750 m. (25 to 37 1/2 chains). For $V = 3\sqrt{R}$ or $4\sqrt{R}$, with adequate cant, speeds of 70 to 80 km. (43.5 to 50 miles) per hour, or of 90 to 100 km. (56 to 62 miles) per hour would result. Since the development of a high speed [V = 100 to 120 km. (62 to 75 miles) per hour] is possible only on level lines, efforts should be made to *abolish individual curves of small radius, especially between long straight alignments, by increasing the radius of the curves* to a value corresponding to these speeds.

Between two reverse curves there is usually interposed between the origins of the transition curves *an intermediate straight* section having a *minimum length* of from 10 to 40 m. (33 to 130 feet). For high speeds 50 to 70 m. (164 to 230 feet) are recommended as desirable lengths.

The speed is taken into consideration only indirectly, by the fact that in special cases shorter lengths are authorised for lower speeds.

The obvious connection between the length of the intermediate straight section and the lengths of the adjacent superelevation curves is taken into consideration only in a few cases. The Netherlands Railways, the Swiss Federal Railways and the German State Railways alone consider that *on high-speed lines,*

with the use of long transition curves and gently graded superelevation ramps, an intermediate straight section is unnecessary; the Netherlands Railways apply this rule when the superelevation ramps have double the ordinary length. Under the conditions laid down in the chapter « Superelevation » to which we would refer here, the question of the abolition of the intermediate straight stretch is examined at length. The conclusion to be drawn from this study is that in the case of the use of long transition curves and sufficiently low graded superelevation ramps, its abolition does not offer any inconvenient feature.

Between curves of the same sense the greater number of the administrations *do not interpose an intermediate straight section;* a few others use straight sections 20 to 40 m. (65 to 130 feet) in length. According to the considerations explained in the chapter « Superelevation » it is better to replace this intermediate straight length by an intermediate curve of a greater radius. In this manner the continuity of the change of direction at high speeds is the better ensured.

When *on double track lines one road has to be slewed from the narrow six-foot way of the standard track to the wide spacing in a station, curves having radii* of 300 to 5 000 m. (15 to 250 chains) are used. In this case, however, there are grounds for advocating radii of $R \geqslant 10\,000$ m. (500 chains), recognising that with small angles at the centre of curvature the length of the curve becomes too small and makes itself felt in a disagreeable manner at high speeds by jerks in the train.

When a curve in the open road and one of a turnout join in the same sense, this case is treated by most administrations in a similar manner to that of a compound curve in the running road.

When a curve in the open road and one of a turnout join in opposite sense this case is treated by most administrations in a similar manner to that of *reverse curves in the open road. Only* the Bergslagen Railway makes a distinction, depending upon whether the turnout is run onto in the facing or the trailing direction, and does not interpose a straight section in the second case.

Between the termination of the turnout and a following curve in the open road most administrations also interpose an *intermediate straight section* behind the turnout. No administrations reports that it bases the length of this straight section on the speed.

For maintaining the curvature, *fixed reference marks* are generally used. As required the curves are adjusted by measuring the versines or by similar methods.

On important lines run over by the international express services existing *curves of relatively small radius are usually replaced by others of greater radius* when opportunity offers *in order to increase the safety and the speed.*

To conclude, the following may be said :

High speeds and great safety necessitate curves of as large a radius as possible. Greater importance than that apparently given at present, judging from the particulars received, ought to be attached to the desirability of maintaining as far as possible on curves the maximum speed authorised for the open roads. In many cases this result can be obtained without the use of elaborate means by increasing the radius of the curves. At the same time the economic conditions would be improved. The higher the superelevation selected, the higher may be the speed on curves. The limit is fixed by the amount of superelevation which is practically usable.

Between two reverse curves a straight intermediate length does not appear to be necessary provided that long transition curves and gently graded superelevation ramps assure the transition be-

tween the curve and the reverse curve. Between two curves of the same sense, it is desirable to interpose, instead of an intermediate straight section, an intermediate curve of a greater radius. For widening the six-foot way at station approaches, it is recommended that curves exceeding 10 000 m. (500 chains) radius should be used. With the object of maintaining the exact curvature of lines in the open road, fixed reference marks should be used.

CHAPTER IV.

Superelevation.

A certain superelevation of the outside line of rails, varying, according to the views of the different railway administrations, between wide limits, has been used for many years, principally for economic reasons. About 40 years ago, the centre of gravity of the locomotive as a whole at about 1.30 m. (4 ft. 13/64 in.) above the top of the rail, was relatively low, so that safety did not play an essential part and for this reason with the comparatively low speeds of the period, only a very small superelevation which might even become zero in curves of great radius, was considered necessary. Consequently, for a long time, the economic aspect was the only determining factor in the calculation of the superelevation. This point of view is put forward for example in the « Technical agreements » of the German Railway Association of 1909, section 7, § 3, where it is stated : on curves, account being taken of the radius and the speed, the superelevation ought to be determined in such a manner that the wear of the two lines of rails should be as far as possible the same.

The projected increase of *speeds on curves and the consideration of the stability of locomotives* with a centre of gravity of the engine as a whole placed higher and higher, until it ultimately reached the height of 2.10 m. (6 ft. 10 9/32 in.) above the top of the rail, have led the German Railway Association to *re-investigate the subject of the superelevation on curves.* This question of the superelevations was studied from the point of view : 1. of the wear in depth of the two rails; 2. of safety; 3. of the stability of running; 4. of economic conditions (wear in depth and in the lateral direction). The result of these researches is to be found in the « Technical agreements » of the German Railway Association, 1930 edition, section 30, § 2. On the assumption that the centre of gravity of the whole locomotive will not exceed of 2.00 m. (6 ft. 6 3/4 in.) above the top of the rail, it is recommended, denoting by

h the superelevation in millimetres,

V the maximum speed on the curve in km./h.,

R the radius of the curve in metres, that a superelevation should be chosen which is not greater than $h = 8\,\frac{V^2}{R} + 30$ (in exceptional cases up to $h = 8\,\frac{V^2}{R} + 40$), nor smaller than $h = 8\,\frac{V^2}{R} - 30$. The superelevation should not exceed 150 mm. (5 29/32 inches) nor be less than

$$h = 11.8\,\frac{V^2}{R} - 90$$

It is recommended to make the *superelevation,* between stated limits, *as great as possible.* The more the superelevation approaches the value

$$h = 8\,\frac{V^2}{R} + 30$$

the more the *economic aspect* is safeguarded by the wear in depth of the outer rail and of the inner rail being as uniform as possible, and by the side wear of the outer rail being as little as possible.

On the other hand the superelevation calculated by means of the formula

$$h = 11.8 \frac{V^2}{R} - 90$$

represents the minimum value which still assures sufficient stability of the locomotive, i. e. still assures *safe working*.

It is quite natural that the above formulæ should be compared with the familiar theoretical formula

$$h = 11.8 \frac{V^2}{R}$$

indicating that the centrifugal force is balanced by the superelevation. The maximum values of $\frac{V^2}{R}$ are generally comprised approximately between $\frac{V^2}{R} = 12$ for small radii of curves and $\frac{V^2}{R} = 16$ for large radii of curves.

Since :

$$8 \frac{V^2}{R} + 30 = \left(8 + 30 \frac{R}{V^2}\right) \frac{V^2}{R},$$

$$8 \frac{V^2}{R} - 30 = \left(8 - 30 \frac{R}{V^2}\right) \frac{V^2}{R},$$

$$11.8 \frac{V^2}{R} - 90 = \left(11.8 - 90 \frac{R}{V^2}\right) \frac{V^2}{R},$$

where the quantity in brackets may be considered as representing the coefficient of superelevation u relatively to the theoretical coefficient of superelevation 11.8, we find for

	for $\frac{V^2}{R} = 12$	16
$u = \left(8 + 30 \frac{R}{V^2}\right) =$	10.5	9.9
$u = \left(8 - 30 \frac{R}{V^2}\right) =$	5.5	6.1
$u = \left(11.8 - 90 \frac{R}{V^2}\right) =$	4.3	6.8

The differences therefore decrease in proportion as the value $\frac{V^2}{R}$ increases.

The *main aspects* to be considered in the *determination of the superelevation* have been mentioned by the German State Railways.

These are as follows :

1. Ensuring safety for locomotives with high centres of gravity;

2. Stability of the vehicles while running;

3. Reduction of the side wear of the outer rail.

According to the experience acquired up to the present, these objects are attained if the superelevation is determined by means of the standard formula

$$h = 8 \frac{V^2}{R}$$

in use up to now. The German State Railways are considering increasing the superelevation with the object of reducing still further the lateral wear of the outer rail; on the other hand, it has recently authorised the use of the formula

$$h = 8 \frac{V^2}{R} - 30$$

in exceptional cases for difficult local conditions on curves of small radius. But *in principle the greatest superelevation* allowable under local conditions should always be employed.

The Netherlands Railways, where the superelevation is determined by means of the formula

$$h_{mm} = \frac{2}{3} \cdot \frac{V_{m\,h}}{R_m} = 667 \frac{V_{km\,h}}{R_m}$$

consider that this superelevation is rather too little and have the intention of increasing it with the idea of improving safety and reducing the lateral wear of the outer rail.

The determination of the superelevation is based, on the railways which have replied to our questionnaire, partly on experience, and partly on theoretical

considerations. Most commonly, the superelevation is determined by means of the fundamental formula

$$h_{mm} = c \cdot \frac{V_{km/h}}{R_m}$$

in which the value attributed to C is 500 on the Danish State Railways and the Norwegian State Railways, 667 on the Netherlands Railways, the Swedish State Railways, the Grängesberg-Oxelösunds Railways, the Bergslagen Railway, and 700 on the Swiss Federal Railways.

Putting

$$h = c \cdot \frac{V}{R} = u \cdot \frac{V^2}{R},$$

in order to be able to compare once again the coefficient of superelevation u with the theoretical coefficient 11.8, the values of u indicated in the table hereafter are obtained.

V km. (*miles*) per hour.		40 (25)	42.4 (26.3)	56.5 (35.1)	59.2 (36.8)	60 (37.3)	80 (50)	100 (62)	120 (75)
u	$\frac{500}{V}$	12.50	11.80			8.33	6.25	5.00	4.17
	$\frac{667}{V}$	16.93		11.80		11.12	8.46	6.67	5.56
	$\frac{700}{V}$	17.50			11.80	11.67	8.75	7.00	5.83

This table shows that at low speeds the value of u exceeds 11.8, while it diminishes more and more as the speeds increase, so that at high speeds the excess of unabsorbed centrifugal force constantly increases.

The Dutch East Indies State Railways apply the theoretical superelevation formula which gives for the gauge s = 1.067 m. (3 ft. 6 in.) the superelevation $h_{mm} = 8.4 \frac{V^2_{km/h}}{R_m}$ which corresponds to the theoretical value $h_{mm} = 11.8 \frac{V^2_{km./h.}}{R_m}$ which is obtained for standard gauge track s = 1.435 m. (4 ft. 8 1/2 in.). But since a lower mean speed is introduced for V^2, so that on the average V^2 = approx. 0.6 V^2_{max}, the superelevation becomes

$$h_{mm} = \text{approx. } 5.0 \frac{V^2_{km\,/h.}}{R_m}$$

corresponding to a superelevation applicable to the standard gauge track

$$h_{mm} = \text{approx. } 7.1 \frac{V^2_{km./h.}}{R_m}$$

The Portuguese Railways start with the theoretical formula

$$h_m = \frac{s_m \cdot v^2_{m\,/sec^2}}{g_{m./sec.^2} \cdot R_m}$$

and finish, after substitution of the values s = 1.665 m. (5 ft. 5 35/64 in.) for the gauge and after solving $V^2_{m./sec.^2} = \frac{V^2_{km^2/h^2}}{12.96}$ on the assumption of a mean speed V = 75 km. (46.6 miles) an hour, or $\frac{75.V}{12.96}$ with the superelevation formula

$$h_m = \frac{1.665 \cdot 75 \cdot V}{9.81 \cdot 12.96 R} = \text{approx. } \frac{V}{R},$$

which, for standard gauge, would take the form

$$h_{mm} = 886 \frac{V_{km./h.}}{R_m}$$

Most frequently the value adopted for V in the *superelevation formula* and especially in the fundamental formula

$$h_{mm} = c \cdot \frac{V_{km./h.}}{R_m}$$

is the *maximum speed laid down* in the time tables, and more rarely a mean speed (Finnish State Railways, Dutch East Indies State Railways, Portuguese Railways and the Madrid-Saragossa-Alicante Railways).

A *reduced superelevation* is frequently employed in stations, in cases where goods traffic predominates (Swedish State Railways and Bergslagen Railway) or on curves of short length (German State Railways).

The *tolerance* allowed for the superelevation is comprised between the limits of ± 3 mm. to ± 10 mm. (± 15/128 in. to ± 25/64 in.). In this connection a special arrangement on the Dutch East Indies State Railways is worthy of note : two points are marked opposite one another on each of the rails of a curve. and when a plane is made to pass through three points the fourth point must not be higher up by more than 7 mm. (9/32 in.) for V > 45 km. (28 miles) per hour, nor more than 10 mm. (25/64 inch) for V < 45 km. per hour. The distance between the pairs of points not being specified, we have no base on which to ascertain the authorised degree of torsion of the surface.

The superelevation is obtained almost generally by *raising the outer rail;* the Swiss Federal Railways and the Portuguese Railways are alone in raising the outer rail and lowering the inner rail, each by half the superelevation figure.

If *a proportion of the express trains run through the station without stopping, the superelevation* is usually determined, so far as possible, *as for a running road;* the Swiss Federal Railways alone reduce it to 2/3 of the normal figure.

In *compound curves* — adjacent curves of different radii — the difference is compensated in the superelevations by superelevation ramps, either half on each side of the tangent point of the curves, or entirely in the curve having the greatest radius. If it is realised that, in the distance equal to the total wheel base of a vehicle, the transition occurs of the unabsorbed centrifugal force of one curve to that of the adjacent curve, it appears worth while to start the *superelevation ramp* at the tangent point, for one reason to avoid an unloading of the outer guiding wheel, and in consequence the oscillations of the vehicle, and for another reason, to obtain smooth running, and to maintain it in the *curve of smaller radius.*

The superelevation is generally continued *in the short straight sections between curves of the same sense.* The Portuguese Railway Company fixes the superelevation of the straight section between two adjacent curves of the same radius at half that for these curves; if the adjacent curves have different radii, it is equal to half the smallest superelevation. Now, a short straight superelevated section between two adjacent superelevated curves creates a very unfavourable situation from the point of view of safety and also of smoothness of running, as the following considerations demonstrate :

Let G be the weight carried by a pair of wheels, in kgr. :

V the speed, in km. per hour,

v the speed, in metres per second $\left(= \frac{V}{3.6}\right)$,

R the radius of the curve, in metres,

g the acceleration of gravity = 9.81 m. per sec^2,

C the centrifugal force imposed upon a pair of wheels in kgr., or

$$C = \frac{G}{g} \cdot \frac{v^2}{R} = \frac{G\,V^2}{127\,R} \text{ in kgr.}$$

s the distance between planes through the tread circles, in metres,

h the superelevation, in metres.

and with the object of simplification, let us suppose the curves adjacent to the straight stretch have the same radius R. Then a pair of wheels is subjected :

1. in the first curve to the excess centrifugal force

$$\Delta C_r = G\left(\frac{V^2}{127\,R} - \frac{h}{s}\right)$$

2. in the intermediate straight section since $R = \infty$ to the lateral force

$$\Delta C_g = -\,G\,\frac{h}{s}$$

3. in the second curve to the excess centrifugal force

$$\Delta C_r = G\left(\frac{V^2}{127\,R} - \frac{h}{s}\right).$$

On a short run, i. e. in a shorter and shorter time according as the speed increases, the positive value ΔC_r becomes the negative value ΔC_g , then again the positive value ΔC_r . We will now show by an example what are the numerical values in question. For G = 16 000 kgr., V = 110 km. an hour, R = 800 m., $\frac{V^2}{R} = 15.13$, $h = 0.12$ m., $s = 1.50$ m. we get

$$\Delta C_r = 16\,000 \cdot \left(\frac{15.13}{127} - \frac{0.120}{1.500}\right) = 640 \text{ kgr.}$$

$$\Delta C_g = 16\,000 \cdot \frac{0.120}{1.500} = -\,1\,280 \text{ kgr.}$$

If it is in addition assumed that the intermediate straight stretch is $l = 20$ m. and is therefore traversed in the time

$$t = \frac{3.6 \cdot l}{V} = \frac{3.6 \cdot 20}{110} = 0.65 \text{ sec.}$$

it is evident that the variation in force undergone in so short a time from $\Delta C_r = 640$ kgr. to $\Delta C_g = -\,1\,280$ kgr. and from $\Delta C_g = -\,1\,280$ kgr. to $\Delta C_r = 640$ kgr. must exercise an extremely prejudicial action on safety and smoothness of running, even taking into account the fact that in reality the total wheel base of the vehicle and the time corresponding to the displacement which this wheel base represents attenuate the effects of the forces to some small extent. These considerations show the imperative need of eliminating *short straight sections between curves of the same sense*. With this object, we may interpose between the curves of radii R_1 and R_2 a curve of radius $R_3 = R_1 + R_2$ and prolong the superelevation, starting from the two tangent points of the adjacent curves, into the curves of radii R_1 and R_2. A still better solution consists in employing partial transition curves for passing from radius R_1 to radius R_3 and from the latter to radius R_2, and making the intermediate superelevation ramps coincide with these. Thanks to the replacement of the intermediate straight section by the *intermediate curve*, a prejudicial unloading of the outside guiding wheel is avoided in any case and as a result, the oscillations of the vehicle.

Between *two reverse curves*, an intermediate straight section is usually interposed, having a length of at least 10 to 40 m. (33 to 130 feet).

If the historical evolution of the intermediate straight length be followed, it will be seen that it was originally used only between circular curves in order to reserve the necessary space for the possible subsequent introduction of following transition curves. Later, after the adoption of transition curves with corresponding superelevation gradients,

there was provided a straight intermediate section the length of which represented at least the greatest wheel base of the vehicles in service. The object of this arrangement was to avoid the risk of the extreme axle of a vehicle embracing reverse curves, while passing the point of inversion even when the radius was very great. The length of this straight section situate between the transition curves was fixed by the recommended practice of the German Railway Association first at 10 m. (33 feet) and later at 30 m. (98 1/2 feet). If it is remembered that in all countries, especially on mountain lines, circular curves are still frequently connected with straight alignments without transition curves, so that the passage from the curve to the straight takes place during the transit of the total wheel base of a vehicle, i. e. in a very short time, it is obvious that the passage of the same vehicle when its centre traverses the point of junction from the transition curve of much greater radius to the reverse transition curve of equal radius, must be more favourable in all circumstances. Let us explain what happens, by means of an example. If a circular curve of radius $R = 300$ m. is connected without a transition curve to a straight alignment and if the greatest wheel base to be dealt with is $a = 12$ m., the transit of $\rho = R = 300$ m. to $\rho = \infty$ is achieved on the span $a = 12$ m., and at the speed $V = 65$ km. per hour in the time

$$t = \frac{3.6\,a}{V} = \frac{3.6 \cdot 12}{65} = 0.67 \text{ sec.}$$

But if two reverse curves, each of $R = 300$ m. are united by transition curves, each of $l = 40$ m. and if one considers the instant when the middle of the vehicle, assumed symmetrical, is located at the junction point, the extreme pairs of wheels of the vehicle are each at a distance from this point equal to $\frac{a}{2} = 6$ m. and advance momentarily on opposite curves of radius $\rho = \frac{300 \cdot 40}{6} = 2\,000$ m. Now this movement of rotation corresponds to a supposed transition of the half radius $\frac{\rho}{2} = 1\,000$ m. to $\rho = \infty$. The distance a and the time t remain the same, but instead of a radius of curvature of 300 m. we have put one of 1 000 m. so that the passage of the vehicle from the curve of $R = 300$ m. without transition curve, into the straight, is more unfavourable in the ratio $\frac{1\,000}{300} = 3.33$ times that of its passage from a transition curve to the opposite curve. If in consequence, the rotation of the vehicle about the vertical axis does not give rise to valid objections, the rotation about the horizontal axis, in all the circumstances, becomes much more favourable from the fact that it operates uniformly and is not disturbed by the straight intermediate section. Many well known competent authorities [Bräuning, Petersen, Dr. Schwanter and Chappellet (1)] have consequently decided in favour of the *elimination of the straight intermediate section* between reverse curves, if care is taken to provide transition curves of as great a length as possible and *superelevation inclines* of easy gradient.

At *level crossings* of important roads carrying automobile traffic, the superelevation on curves of double track lines is generally carried out in such a way that the *inner rails are at the same height.* It is only in a few cases that a reduced superelevation is used.

One of the principal objects of the superelevation is *to neutralise as far as possible the action of the centrifugal force of trains travelling at high speeds.* According to the particulars furnished

(1) J. CHAPPELLET : « Diagrams of versines and superelevation », the *Bulletin of Railway Congress*, October 1930.

by the railways who have replied to the questionnaire, it is usually attained with the ordinary superelevations, the exceptions being the Finnish State Railways and the Madrid-Saragossa-Alicante Railways. But as these two administrations introduce a mean speed into the superelevation formula, it is necessary according, to our previous remarks, that the coefficient of superelevation $u = \frac{R}{V^2}$ should become smaller and smaller at high speeds, relatively to the theoretical coefficient of superelevation of 11.8.

We will therefore examine under the following circumstances, the conditions created by the *choice of the superelevation, from the point of view of smoothness of running.* The action of the centrifugal force is completely neutralised if the superelevation is determined from the equation

$$h_{max} = \frac{s \cdot v^2_{max}}{g \cdot R} = \frac{s}{g} \cdot p_{max}$$

in which

h_{max} is the superelevation in metres, when the centrifugal force is completely balanced,

s the distance apart of the planes passing through the tread circles, in metres (for standard track $s = 1.50$ m.),

v_{max} the maximum speed in metres per second,

$$\text{and } v = \frac{V}{3.6}, \text{ if}$$

V is the speed in km. per hour,

g the acceleration due to gravity $= 9.81$ m. per sec^2,

R the radius of the curve in metres,

$$p_{max} = \frac{v^2_{max}}{R} = \frac{V^2_{max}}{12.96\,R}$$

the normal acceleration on the curve in m. per sec^2;

We have for the standard track :

$$p_{max} = h_{max} \cdot \frac{g}{s} = h_{max} \cdot \frac{9.81}{1.500} = 6.54\, h_{max}$$

and if h_{max} is expressed in mm. :

$$p_{max} = \frac{6.54}{1\,000} = \frac{h_{max}}{153} \text{ or } h_{max} = 153 p_{max}$$

For any superelevation whatever, we then have :

$$p = \frac{h}{153} \text{ and } h = 153\, p.$$

In consequence *the amount of the superelevation h may serve to measure the part of the acceleration balanced out* p . When the speed reaches its maximum allowable value, h is generally smaller than h_{max}. The difference h_Δ is expressed by the equation

$$h = h_{max} - h_\Delta = 153\,(p_{max} - p_\Delta),$$

from which the effective acceleration is deduced :

$$p_\Delta = p_{max} - \frac{h}{153}.$$

and for the standard track :

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{h}{153}.$$

The more the value h of the superelevation is reduced, the more the effective acceleration p_Δ increases.

A comparison between different values of superelevation used for the same radius of curve R and the same speed V will explain the change of p_Δ which results therefrom. Let R = 500 m. (25 chains) and V = 80 m. (50 miles) an hour. In this case, the Finnish State Railways take as a basis an average speed $V_m = 60$ km. (37.3 miles) an hour, giving a superelevation of

$$h = 600 \cdot \frac{V_m}{R} = 600 \cdot \frac{60}{500} = 72 \text{ mm.}$$

On putting V = 80 km. an hour, the superelevation h would then be given by the equation

$$h = 450 \cdot \frac{V}{R} = 50 \cdot \frac{80}{500} = 72 \text{ mm.} \left(2\frac{53}{64} \text{ in.}\right)$$

When $\frac{V^2}{R} = \frac{6\,400}{500} = 12.8$, we find

$$p_\Delta = \frac{12.8}{12.96} - \frac{72}{153} = 0.99 - 0.47 = 0.52 \text{ m.}$$

(1.70 ft. per sec².).

The German State Railways use the formula

$$h = 8\,\frac{V^2}{R}$$

whence is obtained, under the same assumptions,

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{8\,V^2}{153\,R}$$

$$= \frac{12\,8}{12\,96} - \frac{102.4}{153} = 0.99 - 0.67 = 0.32 \text{ m.}$$

(1.05 feet per sec².).

The observations made and the results obtained, namely in braking trials have led to the adoption of the rule that the acceleration should not exceed

$$p_\Delta \leqq 0.40 \text{ (1.31 ft.) per sec}^2$$

to produce smooth running. If we put

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{h}{153} = 0.40,$$

the value of the superelevation

$$h = \frac{153\,V^2}{12.96} - 153 \cdot 0\,40 = 11.8\,\frac{V^2}{R} - 61$$

may be deduced therefrom.

On the German State Railways the values of $\frac{V^2_{max}}{R}$ are comprised, for R = 300 m. (15 chains) to R = 1 000 m. (50 chains), between $\frac{V^2_{max}}{R} = 14$ to 15. If we take for our basis $\frac{V^2_{max}}{R} = 15$, we have

$$h = 11.8 \times 15 - 61 = \text{approx. } 8 \cdot 15$$

or in a general manner,

$$h = 8\,\frac{V^2}{R}.$$

The minimum superelevation given in the « Technical Agreements » of the German Railway Association

$$h = 11.8\,\frac{V^2_{max}}{R} - 90$$

would therefore correspond to an acceleration

$$p_\Delta = \frac{h_\Delta}{153} = \frac{90}{153} = \text{env. } 0.60 \text{ m/sec}^2.$$

The general equation

$$p_\Delta = \frac{V^2}{12\,96\,R} - \frac{h}{153}$$

also enlightens us on the question of knowing how p_Δ can undergo, in practice, noteworthy increases by the excess values of V, by the reductions of R and of h and consequently sensibly detract from smoothness of running and sometimes imperil the safety of working. Since for p = about 2 m. (6.56 ft.) per sec² derailment towards the inside is to be feared, it is necessary to ensure, for reasons of safety that the value p_Δ remains as far as possible below this limit. To obtain a high degree of safety and above all smooth running, a superelevation as great as possible must be adopted, even at the cost of unavoidable modifications of the curves as regards level and curvature. Thus for example the very smooth running on the London and North Eastern Railway at high speeds [V = 120 to 132 km. (75 to 82 miles) an hour] is attributable, among other things, to the superelevation used on that system, namely

$$h = 10.2\,\frac{V^2_{max}}{R}$$

which gives, using the values usual on that line, viz. $\frac{V^2_{max}}{R} = 12.96$ to — in exceptional cases — 19.7, the very low figure $p_\Delta = 0.13$ to 0.20 m. (0.426 to 0.656 ft.) per sec².

For turnouts, the value $p_\Delta = 0.40$ m. (1.31 ft.) per sec² is exceeded on nearly all railways with the object of obtaining as high a speed as possible, because in this case the four rails resting upon long sleepers offer more resistance to a change of level and of curvature of the junction lead. It goes without saying that an increase of p_Δ adversely affects the quality of running. But since a fast train rarely has to pass over, on a long journey, the leads of junctions, these exceptions may be allowed so long as the running remains within tolerable limits and provided above all that the maximum authorised speed is not exceeded.

In *curves of short length* in the running road, the majority of the administrations who replied to the questionnaire *reduce the superelevation,* which has the effect of raising the value p_Δ as in the case of *turnouts.* The other administrations use the full superelevation, even in curves of short length. But this condition can only be fulfilled by the continuation of the gently graded superelevation ramps into the straight lengths or by increasing the grade of the superelevation ramps. In the first case, it is smoothness of running which is achieved; in the second case it is safety. The German State Railways do not provide superelevation for short curves of the track between turnouts, but when necessary require the speed to be reduced to the desired extent; on the other hand, they attach importance to the *interposition of transition curves, no matter how short they may be.*

The result of the foregoing discussions on the subject of superelevation, may be summarised as follows :

The decisive factors for the determination of the superelevation are :

1. *the safety of the locomotives with high centres of gravity at high speeds;*

2. *the smoothness of running of the vehicles;*

3. *the good economic result obtained by the reduction of side wear of the outer rail.*

Researches carried out as to the amount of superelevation which best meets these conditions on standard gauge tracks have shown that the economic aspect is the better safeguarded as the figure for the superelevation approaches nearer to the value

$$h = 8\frac{V^2}{R} + 30,$$

while in exceptional cases it may be reduced to

$$h = 8\frac{V^2}{R} - 30.$$

For reasons of safety it is desirable that the superelevation figure should not fall below

$$h = 11.8\frac{V^2}{R} - 90$$

In this manner the superelevation has become a function of the ratio $\frac{V^2}{R}$ *i. e. of the normal acceleration, while formerly most of the formulæ for the superelevation were based on the ratio* $\frac{V}{R}$. *For the calculation of the superelevation, the maximum speed given in the time tables will be adopted as the value for* V. *The superelevation figure may serve to measure the proportion of the acceleration neutralised. The effective acceleration for the standard gauge track is*

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{h}{153}.$$

If the superelevation is continued into a straight alignment the first term of the second part of the equation disappears and there would be produced at high speeds an inversion of the acceleration and, arising therefrom, of the force, which shows itself in an un-

favourable light by the unloading of the guiding outside wheel. The more the superelevation figure h decreases, the more the effective acceleration p_Δ *increases. It increases uniformly if the transition curve and the superelevation ramp coincide or if, in default of a superelevation, a transition curve has been interposed. To realise smooth running at high speeds it is necessary in general that we should have*

$$p_\Delta \leqslant 0 \text{ m. } 40 \text{ m. } (1.31 \text{ ft.}) \text{ per sec}^2,$$

while, in exceptional cases especially in curves without superelevation, of the track and of turnouts,

$$p_\Delta = \text{approv. } 0.60 \text{ m. } (1.97 \text{ ft.}) \text{ per sec}^2$$

can ordinarily be considered as the limiting value for sufficiently smooth and safe running. In practical service, allowance must be made for the fact that p_Δ *may increase as the result of* V *being exceeded and of the diminution of* R *and h. The smoothness of running and, in certain circumstances, the degree of security are affected thereby.*

CHAPTER V.

V. — Transition curves and superelevation inclines.

Between a straight alignment and a circular curve a transition curve is interposed in order to permit of the *progressive change of the angular speed of rotation* of a vehicle about a vertical axis when it passes from the straight to the circular curve or from the circular curve to the straight. A transition curve which fulfils approximately the condition of a uniformly increasing angular speed of rotation or of a uniform acceleration of rotation is the cubic parabola. During the displacement along the transition curve and during the time which corresponds to this displacement, the normal acceleration increases uniformly from 0 to p_Δ. A measure of this variation of the normal acceleration is furnished by the derivative expressing the acceleration as a function of the time

$$\Psi = \frac{dp}{dt} \text{ in metres per sec}^3,$$

if we denote, in a general manner, by p the normal acceleration in metres per sec^2, and by t the time in seconds. The value of p with which we are here concerned is represented, in the preceding chapter « Superelevation » by the equation

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{h}{153}.$$

If the transition curve has a length l in metres, it is traversed at the uniform speed V in km. per hour in the time

$$t = \frac{3.6\,l}{V}.$$

Consequently we obtain for the variation of the acceleration the value

$$\Psi = \frac{p_\Delta}{t} = \frac{\left(\frac{V^2}{12.96\,R} - \frac{h}{153}\right)V}{3.6\,l},$$

which in the first instance is applicable only to an individual pair of wheels. For a vehicle whose total wheel base is a in metres the transit is performed in the displacement $l + a$ in the time

$$t = \frac{3.6\,(l+a)}{V}.$$

In this case we then have

$$\Psi = \frac{p_\Delta}{t} = \frac{\left(\frac{V^2}{12.96\,R} - \frac{h}{153}\right)V}{3.6\,(l+a)}.$$

If no transition curve l exists, nor in default of a superelevation *h*, a superelevation ramp, we have

$$\Psi = \frac{V^3}{46.66\,Ra}$$

a value which is appreciably greater as a result of the increase of the expression of the numerator and of the decrease in the denominator. In order to have an idea of the numerical magnitude of Ψ we will compare for a vehicle whose overall wheel base is $a = 15$ m. (49 ft. 2 1/2 in.):

1. a curve of the running road having a radius R = 800 m. (40 chains), the superelevation $h = 120$ mm. (4 23/32 inches) and the transition curve of length $l = 130$ m. (427 feet), with the speed V = 110 km. (68.3 miles) per hour, and subsequently;

2. a curve of a turnout having a radius R = 190 m. (9 1/2 chains) with the speed V = 45 km. (28 miles) per hour.

We get:

for the first case $\Psi = 0.082$ m. (0.269 ft.) per sec^3,

for the second case $\Psi = 0.685$ m. (2.246 ft.) per sec^3.

The extremely great difference between the numerical values of Ψ characterises *the great importance of a long transition curve* from the point of view of smoothness of running and of safety.

Another interesting interpretation might be given to the equation

$$\Psi = \frac{p_\Delta}{t}.$$

We have already drawn attention to the fact that in the curves the effective acceleration p may increase sensibly as a result of exceeding V and of reducing R and h. On the straight, however, in the event of variations in level accelerations may also be set up, of the value

$$p_\Delta = -\frac{h}{153}$$

where the negative sign indicates only the direction. In each case, whether on the straight or on the curve, the variation of p_d takes place over a displacement s. If we then designate by p_d the difference between the initial value p_Δ and the value p_Δ which develops along the journeys s, and if we introduce it into the equation for Ψ the latter takes the form

$$\Psi = \frac{p_d \, V}{3.6\, s}.$$

This equation makes it possible to determine numerically all the irregularities in the laying of the track under the headings of curvature and level, and from it to deduce conclusions relative to the values of Ψ which must not be exceeded.

All the railway administrations have come to recognise the *importance of the transition curve the quality of which is measured by the value* Ψ. They use in quite a general degree the cubic parabola having for its equation

$$y = \frac{x^3}{6\,C}$$

where $C = R \cdot l$, with R denoting the radius of the curve in metres and l the length of the transition curve in metres. The value C which varies on the different railways between very wide limits, depends essentially on the length l of the superelevation ramp, coinciding more often than not with the transition curve and depending in its turn on the superelevation and the gradient.

The *superelevation ramp* is usually traced *as a straight line.*

For the changes of gradient at the origin and the termination of the superelevation ramp, no definite radii are prescribed, on account of the slight grade of the inclines. The North of Spain Railways compensate the changes of grade by means of short ramps having a gentle slope.

The *gradient of the superelevation ramps* is generally comprised between 3.3 and 1 mm. per m. (between 1 in 300 and 1 in 1 000). Some of the admin-

istrations take account of the speed by prescribing flatter gradients when the speed increases and by thus obtaining gradients of 1 mm. per m. (1 in 1 000). We consider as very practical the proposition already mentioned in the preamble, which appears in the « Technical agreements », section 20, § 3, of the German Railway Association; this proposition tends to *give compulsory effect to the requirement that the gradient of the superelevation ramp should depend upon the speed* in such a manner that the vehicle always takes *the same time* for its rotation about the horizontal axis, which gives the relationship

$$\frac{1}{n} \leq \frac{1}{10\,V}.$$

In most cases the length of the transition curves is made equal to the length of the superelevation ramps, so that the rotations of the vehicle about the vertical axis and the horizontal axis are effected simultaneously. Occasionally to produce a gentle gradient a greater length is given to the superelevation ramp than to the transition curve.

If then the *transition curve is shorter than the superelevation ramp,* recourse is had to one of two solutions concerning their relative position. Some administrations made the extremity of the transition curve coincide with that of the superelevation ramp, so that the latter continues into the straight alignment; others join the origins, so that the superelevation ramp is prolonged into the circular curve. We will return to the considerations bearing upon the subject of short straight lengths between curves of the same sense. In both cases the following effects are produced.

In the first case, at the origin of the transition curve there is a superelevation h_a which is smaller than the superelevation h applied in the circular curve. For an individual pair of wheels a lateral force is exerted at this position equal to

$$\Delta C_g = -\,G\,\frac{h_a}{s},$$

while at the extremity of the superelevation ramp and of the transition curve it is the lateral force

$$\Delta C_g = G\left(\frac{V^2}{127\,R} - \frac{h}{s}\right)$$

which acts. In travelling over the transition curve, there is produced therefore a *reversal of the force* with a change of acceleration Ψ of corresponding intensity which sets up oscillations of the vehicles, especially at the passage from the curve to the straight alignment.

In the second case, on the contrary, there is at the extremity of the transition curve a superelevation h_a smaller than h applied in the circular curve. At the end of the transition curve, the lateral force reaches its maximum value

$$\Delta C_r = G\left(\frac{V^2}{127\,R} - \frac{h_a}{s}\right),$$

at the end of the superelevation ramp, it has a rather smaller value

$$\Delta C_r = G\left(\frac{V^2}{127\,R} - \frac{h}{s}\right),$$

so that on the difference in the distance travelled between the superelevation incline and the transition curve is only

$$G\,\frac{h_a}{s} \quad \text{alters to} \quad G\,\frac{h}{s}.$$

The lateral force ΔC_r *thus varies a little* in magnitude but *continues to act in the same sense,* while the variation in the acceleration becomes very appreciably smaller.

In the course of a very great number of trial trips, we have been enabled to observe the oscillations of the vehicles set up by the action of the above-mentioned forces through the continuation

of the superelevation ramp into the straight alignment, while the prolongation of the superelevation ramp into the circular curve does not expose the vehicles to any kind of oscillations. If on entering the curve, the vehicles are leaning towards the outside, they remain in all probability in this position until their departure from the curve. But as the oscillations of vehicles are always accompanied by unloading of the springs, which may even lead to derailments, particularly in the event of the maximum speed being exceeded, due to the unloading of the outer leading wheel, it appears necessary, in principle, to *prolong into the circular curve a superelevation ramp which is longer than the transition curve.*

In brief, we may say that : *the transition curve ought to produce the progressive change of the angular speed of rotation of a vehicle. On the cubic parabola this rotation takes place with uniform acceleration. The change of acceleration*

$$w = \frac{p_\Delta}{t} = \frac{\left(\frac{V^2}{12.96\,R} - \frac{h}{153}\right) V}{3.6\,(l + a)},$$

on which the length l *of the transition curve and the total wheel base* a *of a vehicle exercises, all other circumstances being the same, a decisive influence, which serves as a measure of the quality of a transition curve. The superelevation ramp is generally traced on the straight. The gradient of the superelevation ramp ought to be made to depend in such a manner upon the speed that the vehicle always has the same time available for its rotation. We then obtain an inclination ratio*

$$\frac{1}{n} \leq \frac{1}{10\,V}$$

If the superelevation ramp is longer than the transition curve, it is necessary to extend it into the circular curve.

CHAPTER VI.

Points and crossings.

The *curves of turnouts* differ from the curves in the open road by that the circular curve is generally joined to the straight alignment *without a transition curve.* In a few rare cases only, in order to replace the transition curve up to a certain point, a radius a little greater than that of the turnout lead has been given to the curved tongue, while in front of a straight crossing or behind a curved crossing the straight alignment always connects directly with the turnout lead. It is only when the *turnout lead,* serving as a common route, is situated between the two curves of the running road, that safety can be improved by a superelevation. But if a *turnout lead,* serving as a common route, *which is situated between two straight alignments,* is provided with superelevation, the inevitable prolongation of the superelevation ramp into the straight alignments has for effect *the reversal of the force* studied in the preceding chapter « Transition curves and superelevation ramps » and which is manifested by the unloading of the outer leading wheel and by oscillations of the vehicle. In this case safety is the better ensured if there is no superelevation, at the same time reducing sufficiently the maximum speed authorised elsewhere for a curve of superelevated road. While the curve of the running road, in default of a transition curve, meets the straight alignment tangentially, the turnout on some systems still cuts the straight at a certain angle, and even when turnouts theoretically tangential are used, for constructional reasons a short length of the curve must be replaced by a short straight section whose angle of incidence is, it is true, appreciably smaller. The majority of the administrations still make use of straight crossings with short straight lengths interposed between them

and the turnouts leads. If a running road curve is connected up behind the crossing with a turnout lead having points of the type which form an angle with the stock rail it is inevitable that because of the succession of — straight part — deviation — junction lead — straight part — curve in the open road — the running over such a curve becomes unstable and, at high speeds, dangerous, as a result of the continual change of the curvature $\frac{1}{\rho}$. In consequence, the tangent tongues with a short straight part which strikes the rail at only a very small angle of incidence, and the curved crossings constitute, from the points of view of smoothness of running and of safety, an important improvement, for then the order of succession becomes the following :

Straight part — turnout lead — curve in the running road.

The extension of the turnout lead through the crossing presents, in addition, this advantage that for equal curve radius, the angle of the crossing becomes smaller and that, as a result, the gap in the rolling surface of the outer rail becomes shorter. By comparison with the curve in the open road, *the interruption of the rolling surface of the outer rail,* necessitated by the gap of the crossing, constitutes therefore a drawback, for, no matter how well designed the turnout may be, in the course of time a hollow is formed at the nose of the crossing, by reason of the crushing of the mental, a cavity which produces, by the flattening out of the crossing, worn places on the outer rail leaning towards the crossing.

The result, for leads without superelevation, is a *lowering* of level below normal, and for the others, *a reduction of the superelevation.* Moreover, in the open road curve the outer wheel always leads, while a little in front of the crossing, in the case of a facing turnout, and a little behind the crossing in the case of a trailing turnout, *the guiding effect is transferred from the outer wheel to the inner wheel,* which produces the effect, that for an inclination of 1 in 56 [for R = 190 m. (9 1/2 chains)] and of 1 in 77 [for R = 500 m. (25 chains)] the leading pair of wheels is withdrawn from the outer edge of the rail, interrupted by the groove of the crossing, and brought towards the check rail. For the guiding of the pair of wheels these *inclinations are equivalent to deviations* of 1° 1′ 23″ and 0° 44′ 50″, i. e. to greater angles than those provided in modern turnouts for the angle of incidence or of intersection. The extension of the check rail and the reduction of the inclination at its ends can thus improve appreciably the guiding of the wheels and in certain circumstances raise the limit of speed at which derailment would occur. If with its four rails fastened to continuous sleepers, the turnout lead offers more resistance to changes of level and of curvature than the open road curve, on the contrary it has drawbacks which the latter does not possess : absence of a transition curve, angle of intersection or of incidence of the tongue, interruption of the rolling surface of the outer rail at the crossing, lowering or reduction of the superelevation towards the crossing, guiding effect transferred from the outer wheel to the inner wheel at a small angle and, further, often short sections of rail or short crossing. All these factors produce, on account of the inevitable shocks to which they give rise, disadvantages *as compared with curves in the open* which it is essential to take into account in determining the maximum speeds for turnout leads. The greater the radius of the turnout curve, the more may be diminished the disadvantages mentioned above, excepting the interruption of the rolling surface of the outer rail at the crossing, which becomes longer and the slant of the outer

rail towards the crossing, which is then necessarily more unfavourable. But a greater radius of the turnout curve, if the maximum speed fixed for a smaller radius remains the same, enhances the safety and smoothness of running and in addition increases appreciably the duration of service of a turnout. For these reasons, the majority of administrations consulted have reduced appreciably the maximum speed for turnout curves below that authorised for super-elevated curves in the open road.

From the point of view of high speed, the Netherlands Railways and the Dutch East Indies State Railways make a distinction between the *turnouts in open roads* and *turnouts at station approaches.* The first of these administrations employs mainly symmetrical turnouts with the radius of curve R = 1 200 m. (60 chains) and the angle of the crossing 1 in 15, elsewhere straight points with R = 600 m. (30 chains) and the same angle of the crossing are used; for station approaches in certain circumstances the same turnouts are used, but elsewhere turnouts with R = 300 m. (15 chains) and a crossing angle of 1 in 10 or 1 in 9. The Dutch East Indies State Railways make use in running lines of straight turnouts with the radius of curve R = 400 m. (20 chains) and with a crossing angle of 1 in 14, and for station approaches, turnouts with radius R = 180 m. (9 chains) and crossing angle 1 in 10. The other administrations do not make any difference.

The angle α of intersection or of incidence of the tongue, the *angle β of the crossing,* and corresponding inclination 1 in *n* of the crossing vary between wide limits, as the following table shows :

For	R	α	β	=1 : *n*
	m. (*chains*)	° ′ ″	° ′ ″	
Finnish State	295 (*14 3/4*)	0 41 25	5 11 40	=1 : 11
Madrid-Saragossa-Alicante	300 (*15*)	1 22 31	5 8 34	=1 : 11.12
Portugese Co.	300 (*15*)	1 14 36	5 9 34	=1 : 11.08
Danish State	330 (*16 1/2*)	0 30 0	5 11 40	=1 : 11
Dutch East Indies	400 (*20*)	1 2 30	4 5 8	=1 : 14
Danish State German State	500 (*25*)	0 27 0	4 5 8	=1 : 14
Netherlands	600 (*30*)	0 55 0	3 48 51	=1 : 15
Swedish State	600 (*30*)	0 30 0	3 48 51	=1 : 15

The smallest angle of intersection is therefore α = 0° 27′ 0″, and the smallest crossing angle β = 3° 48′ 51″.

The German State Railways have adopted for their new types of turnouts the following figures :

R	Angle of incidence. α	β	=1 : *n*
m. (*chains*)	° ′ ″	° ′ ″	
500 (*25*)	0 40 0	5 35 59	=1 : 10.2
500 (*15*)	0 33 30	4 20 20	=1 : 13.2

For the *tongues,* the curved form is normally employed; only the North of Spain Railways, the Madrid-Saragossa-Alicante Railways and the Portuguese Railways still use straight tongues. With curved tongues the curves of the points of blades are sometimes replaced by tangents (angle of incidence) and sometimes the tongues cut the rolling surface of the stock rail (angle of intersection).

On some railways the *heel of the tongue* is held by a wooden or steel wedge against the stock rail. The Swiss

Federal Railways enlarge the section of the tongue and of the heel pivot. The German State Railways employ articulated tongues and flexible tongues. The heel of the articulated tongue rests in an articulated member specially designed for this purpose. The other administrations do not take *special* measures to hold the heel of the tongue securely.

The anchorage of the tongue is generally protected against *the action of the braking efforts* by the pivot or the tension of the springs.

Tongues which are housed under the head of the rails are the rule; the old type is no longer in use except on the North of Spain Railways.

While the Dutch East Indies State Railways use only *flexible tongues* by which the groove of the crossing in the common route is closed, the Swiss Federal Railways, the Finnish State Railways, the North of Spain Railways, and the Madrid-Saragossa-Alicante Railways use solely articulated tongues. The two types are in use on the other systems some of which prefer flexible tongues for the main lines.

In *order to ensure that the tongues are home, use is most often* made of a *hook point-locking device or point lock,* serving the purpose of *holding the tongue against the stock rail.* Less frequently use is made of the *toggle joint point lock* which presses the tongue against the stock rail. The Netherlands Railways use a device in which the tongue is lifted by means of a roller upon a double inclined plane resting upon the slide chair and is thrusted, at the bottom of the slope, against the stock rail.

In general the *turnout curve* has a *uniform* development throughout the length of the points. On the Finnish State Railways, the German State Railways and the Bergslagen Railway alone, the tongues have in part a greater radius than the turnout curve.

The *least length of the straight part of the crossing in front of the mathematical point of the nose* is 1 m. (3 ft. 3 3/8 in.) (on the Danish State Railways) and the greatest is 4.74 m. (15 ft. 7 in.) (on the Portuguese Railways). *Behind* the mathematical point of the nose the smallest length is 1.24 m. (4 ft. 53/64 in.) (on the Swiss Federal Railways) and the greatest is 4.327 m. (14 ft. 2 23/64 in.) (on the Swedish Grängesberg-Oxelösund Railways).

The least length of the straight part of the crossing is, according to the replies received, 2.58 m. (8 ft. 5 19/32 in.) (on Swiss Federal Railways) and the greatest is 7.12 m. (23 ft. 4 5/16 in.) (Grängesberg Oxelösund Rys.).

The Danish State Railways, the Netherlands Railways, the German State Railways and the Swedish Bergslagen Railway use for the construction of the turnout curve, *curved crossings.* The radii are 190 m., 300 m. and 500 m. (9 1/2, 15 and 25 chains). With curved crossings, the same gauge is given to the deflected track as for the same curves in the other turnouts.

As a general rule the grooves have the *same widths* in the *through road* and the *turnout road.* In the crossing, the smallest width of groove is 40 mm. (1 37/64 inch) (on the Dutch East Indies State Railways and the Norwegian State Railways) and the greatest is 50 mm. (1 31/32 inches) (on the North of Spain Railways, the Madrid - Saragossa - Alicante Railways and the Portuguese Railways); most commonly the width of the gap is from 44 to 45 mm. (1 47/64 to 1 49/64 inches). At the check rail the smallest width of gap is 40 mm. (1 37/64 inches) (on the Norwegian State Railways) and the greatest is 50 mm. (1 31/32 inches) (on the North of Spain Railways, the Madrid-Saragossa-Alicante Railways and the Portuguese Railways). Most often a width of groove of 41 mm. (1 31/64 inches) is used.

In constructing the crossings, the majority of the administrations *make allow-*

ance for tyres which have reached their limit of wear; a few only have not adopted this practice.

The *wing rail raised* from 4 to 6 mm. (5/32 to 15/64 inch) relatively to the nose of the crossing is to be found only on the Danish State Railways, the Swedish State Railways, the Swedish Bergslagen Railway and the Madrid-Saragossa-Alicante Railways.

Several administrations also use moveable wing rails, controlled by spring pressure.

While on the Norwegian State Railways *crossings built up of rails* are in use exclusively, some of the administrations only use crossings with a *special steel nose* (cast steel, manganese steel or forged steel); the others use both types.

The shortest length of rail in turnouts is 1.50 m. (4 ft. 11 in.) (Portuguese Railways) and the longest 8.25 m. (27 ft. 1 in.) (North of Spain Railways). The *shortest length of crossing* V-piece is 2.25 m. (7 ft. 4 19/32 in.) (Grängesberg-Oxelösund Railways) and the greatest is 5.55 m. (18 ft. 2 33/64 in.) (North of Spain Railways).

In order to avoid rapid wear, the *crossing noses* are made of a *special steel* (most frequently manganese steel), or the nose and wing rails are provided with hardened rolling surfaces. Other measures taken with this object are the housing of the extended nose in the fish plate seat of the wing rail, accompanied by reduction of the width of the gap in the crossing from 49 to 44 mm. (1 59/64 to 1 47/64 inches) (on the German State Railways), the raising of the wing rail or the use of moveable wing rails. As regards permissible wear in height of the nose of the crossing and of the wing rail, maxima given vary between 5 and 15 mm. (25/128 and 19/32 inch). Economic considerations here seem to outweigh safety and smoothness of running.

With few exceptions the administrations also make use of *double curve turnouts*. While in Sweden double curve turnouts springing from the inside of the curve are used only, the other administrations, with the exception of the Swiss Federal Railways and the Finish State Railways make use of double curve turnouts on both the inside and the outside of curves.

For *turnouts* situated in a running road curve with fast trains running over the deviated line the *superelevations* which are the rule, under the same conditions for the through lines are normally used. The Swiss Federal Railways use only 2/3 of the normal superelevation. The North of Spain Railways do not allow any superelevation for turnouts on the outside of a curve, on account of the false superelevation or the lowering of the outer line of rails in the branching off road; the Swedish State Railways are content with mentioning the difficulties which arise at turnouts on the outside of a curve. There are only a few administrations who do not provide any superelevation with turnouts.

As *lateral wear of the rails at double curve turnouts*, two railways alone indicate maximum figures of 3 and 5 mm. (15/128 in. and 13/64 in.). In general there are no fixed requirements in this respect.

The smallest angle adopted for cross-over roads and slip points is comprised between 6° 20′ 25″ (inclination 1 in 9) and 4° 45′ 49″ (inclination 1 in 12). The maximum speed at the turnout, a speed which, it is true, depends only upon the radius of the curve, is from 25 to 40 km. (15.5 to 25 miles) per hour.

As *maximum angle for a cross-over road* the Dutch East Indies State Railways allow $90° = 1 \text{ in } \infty$. On the German State Railways the greatest angle in slip points is 1 in 6.6 and in the running lines it is $85° = 1 \text{ in } 0.0875$.

The other administrations authorise for slip points a maximum inclination of 1 in 7.5 to 1 in 9, with the exception of the Dutch East Indies State Railways

where the greatest inclination is only 1 in 10 = 5°42'38".

Slip points in which the tongues are situated outside the crossing diamond are used by several administrations, for example with a radius R = 180 m. (9 chains) by the Netherlands Railways, with R = 190 m. (9 1/2 chains) by the Norwegian State Railways, the Bergslagen Railway and the German State Railways, with a radius of R = 208 m. (10.4 chains) by the Dutch East Indies State Railways and R = 204 m. (10.2 chains) by the Swiss Federal Railways, and finally with a radius R = 300 m. (15 chains) by the German State Railways.

Slip points with moveable noses at the double crossing are not generally used.

Steel sleepers are only used for turnouts by the Dutch East Indies State Railways, and the Swiss Federal Railways ; the German State Railways also ordinarily use steel sleepers, but use timber ones for the most important lines carrying international traffic and long-distances fast trains. The other administrations make use of *wooden sleepers* exclusively.

The reasons advanced in favour of steel sleepers for turnouts are : rapid and accurate laying, perfect and easy maintenance of the correct gauge, greater safety and better standard of upkeep. Timber sleepers are generally used in turnouts when they are also standard in the running roads. It is argued that they make the running more elastic, and consequently more agreeable. Other reasons mentioned in favour of their use are: the abundance of timber in a country, reduced establishment charges and the possibility of insulating the rails on electrified lines.

For the purpose of deciding upon *the maximum speed through turnouts*, no difference is usually made between connections placed on timber sleepers or on metal sleepers. In the through line, bolted or locked points can nearly always be run over in either the facing or the trailing direction at the maximum speed authorised for the open road. The maximum speed for the curved branching off line is fixed by a number of administrations solely according to the radius of the curve, without concerning themselves about the absence of superelevation. The greater number of the administrations take into account the features described above, namely the curve of the turnout compared with the curve in the open road, by making the maximum speed for the turnout curve sensibly less than that allowed for the curve of the running road. A comparative statement of the maximum speed authorised for turnout curves and the effects thereof will be found in the following table. In order to facilitate the comparison we have calculated and inserted in it the value of the acceleration, investigated in the chapter « Superelevation » :

$$p_\Delta = \frac{V^2}{12.96\ R}$$

and the value of the change of the acceleration examined in the chapter « Transition curves and superelevation inclines »:

$$\Psi = \frac{V^3}{46.66\ Ra}$$

on the assumption of a total wheel base $a = 15$ m. (49 ft. 2 1/2 in.) (for the locomotives or coaches of fast trains), with the assistance of the particulars furnished by the different administrations on the subject of the radii of turnout curves and the speed over these curves.

It will be observed from this table that the Dutch East Indies State Railways reach the highest values for P_Δ and Ψ. The value $V^2 = 13.89$, being very high for a turnout curve without any superelevation, of radius R = 180 m. (9 chains), the degree of stability of

Values of $p\Delta$ and Ψ.

Administrations.	R metres (*chains*)	V_{max} km. (*miles*) per hour	$\frac{V^2_{max}}{R}$	$\frac{V_{max}}{\sqrt{R}}$	$p\Delta = \frac{V^2_{max}}{12.96R}$ m. (*feet*) per sec²	$\Psi = \frac{V^3_{max}}{46.66Ra}$ m. (*feet*) per sec³
Dutch Indies State Rys . .	400 (*20*) 180 (*9*)	70 (*43.5*) 50 (*31*)	12.25 13.89	3.50 3.73	0.95 (*3.12*) 1.07 (*3.51*)	1.23 (*4.03*) 0 99 (*3.25*)
Netherlands Rys	1 200 (*60*) 600 (*30*)	100 (*62*) 75 (*46.6*)	8.33 9.38	2 89 3.06	0.64 (*2.10*) 0 72 (*2.36*)	1.19 (*3.90*) 1.00 (*3.28*)
Bergslagen Ry	245 (*12 1/4*)	50 (*31*)	10.24	3.20	0.80 (*2.62*)	0 74 (*2.43*)
German State Rys	500 (*25*) 190 (*9 1/2*)	60 (*37.3*) 45 (*28*)	7.20 10.66	2.68 3.26	0 56 (*1.84*) 0.82 (*2.69*)	0 62 (*2.03*) 0.68 (*2.23*)
Danish State Rys	500 (*25*) 330 (*16 1/2*)	60 (*37.3*) 45 (*28*)	7.20 6 14	2.68 2 48	0.56 (*1.84*) 0 47 (*1.54*)	0.62 (*2.03*) 0 40 (*1.31*)
Norwegian State Rys . . .	190 (*9 1/2*)	40 (*25*)	8.42	2.90	0.65 (*2.13*)	0.48 (*1.57*)
Finnish State Rys	295 (*14 3/4*)	35 (*21.7*)	4.15	2.04	0.32 (*1.05*)	0.21 (*0.69*)
North of Spain Rys . . .	230 (*11 1/2*)	20 (*12.4*)	1.74	1.32	0.13 (*0.43*)	0.05 (*0.16*)

the compound four-cylinder 4-6-2 locomotive, series 24, in which the centre line of the boiler is situated at 2.30 m. (7 ft. 6 9/16 in.) above rail level (cf. the first part, « Vehicles, Chapter II : Height of the centre of gravity and stability of the locomotive ») drops to the figure of $n = 4$ approximately. The absence of any superelevation and of a transition curve must therefore sensibly detract from the safety of running of the locomotive as well as from the comfort of the journey in the carriage. In this connection we return to the very complete researches of Dr. Uebelacker (1) who has investigated, as an example, the running of a 4-6-0 express locomotive on a curve of radius R = 300 m. (15 chains) and with a superelevation $h = 0.10$ m. (3 15/16 in.), without transition curve. The result of this investigation is summarised as follows : « The maximum speed of V = 65 km. (40.4 miles) per hour for the entry of a locomotive into a curve on the running road [of radius R = 300 m. (15 chains)] without transition curve, may therefore be regarded as very high. It appears singular also that in the regulations relating to maximum speeds authorised on curves, no difference is made between the existence or the absence of a transition curve. Now, the measure of the degree of suddenness of the change of direction at the entry to a circular curve, and above all at the exit from a circular curve without a transition curve and superelevation is given by Ψ which, for a total wheel base $a = 9$ m. (29 ft. 6 3/8 in.) approximate-

(1) Dr. Uebelacker : Über die Massewirkungen bei plötzlichen Richtungsänderungen im Lauf von Eisenbahnfahrzeugen » (The effects of mass when sudden changes of direction occur in the running of railway vehicles), *Organ für die Fortschritte des Eisenbahnwesens*, No. 11, 1st June 1930.

ly, of the 4-6-0 express locomotive, would reach a value

$$\Psi = \frac{V^3}{46\ 66\ Ra} = \frac{65^3}{46.66 \cdot 300 \cdot 9}$$
$$= 2.18 \text{ m. (7.15 feet) per sec}^3.$$

The same value would be obtained, for a superelevation of h=100 mm. (3 15/16 inches), from the equation

$$\Psi = \frac{p_\Delta \cdot V}{3.6 \cdot c \cdot a} = \frac{\left(\frac{V^2}{12\ 96\ R} - \frac{h}{153}\right) \cdot V}{3.6 \cdot \left(1 - \frac{h}{153} \cdot \frac{12.96\ R}{V}\right) a}$$
$$= \frac{0\ 434 \cdot 65}{3.6 \cdot 0.4 \cdot 9.0} = 2.18 \text{ m/sec}^3$$

on denoting by c the ratio of

$$\frac{V^2}{12.96\ R} - \frac{h}{153} \quad \text{to} \quad \frac{V^2}{12.96\ R}$$

which takes account of the change of acceleration. On the assumption of a total wheelbase of the locomotive of a = 9 m. (29 ft. 6 3/8 in.) approximately, the maximum value of Ψ in the appended table becomes

$$\Psi = \frac{p_\Delta \cdot V}{3.6 \cdot a} = \frac{0.95 \cdot 70}{3.6 \cdot 9.0}$$
$$= 2.05 \text{ m. (6.72 feet) per sec}^3$$

and therefore has a value only slightly below that obtained in Dr. Uebelacker's numerical example. The maximum admissible speed for turnout curves without either transition curves or superelevation may be computed with the aid of the particulars given above so long as the data have been fixed relatively to the value ψ, and to the smallest overall wheelbase a of the express locomotives. The equation to be applied is then

$$V_{max} = 3.6 \sqrt[3]{R \cdot a \cdot \Psi}.$$

The results of any calculations and experiments which may have been made on the subject of this important maximum value of Ψ, on which depends safety of running over curves, have not, so far as is known, been made available up to the present. However, the interpretation of certain experimental results enables the conclusion to be drawn that it would not be desirable to exceed the maximum value of Ψ = 2 m. (6.56 ft.) per sec^3.

As a means of increasing the speed over turnout curves the gradual increase of the radius of the curve is recommended above all, but the difficulties presented by the construction of long tongues and small inclines for crossings have not yet been solved. In addition, we consider as being pertinent to this object a good lateral support of the point rail against the stock rail with the use of stops on the latter, the adoption of gaps as narrow as possible and the more rigid anchoring of the stock rails.

The preceding considerations may be summarised as follows :

The turnout curve is inferior to the curve of the running road. It is true that by its foundation it is able to offer a greater resistance to changes of level and of curvature than the curve of the open road, but several objections arise to counterbalance this advantage : absence of a transition curve, angle of incidence, or of intersection of the crossings, interruption of the rolling surface of the outer rail at the nose of crossing, lowering or reduction of the superelevation on the side of the crossing, guiding effect transferred from the outer wheel to the inner wheel at a small change of angle, finally multiplication of the points of shock due to the short sections of rail or to crossings of small length. The use of tangent tongues with a small angle of incidence and the prolongation of the junction curve through the crossing, with narrowing of the grooves, constitute important improvements. An extended stock rail, with entry ramps inclined as gently as possible, may sensibly improve the guid-

ing of the leading pair of wheels and make it safer. Since the speed must be less over turnout curves, without either superelevation or transition curves, owing to the increased acceleration $p\Delta$ and the change of acceleration Ψ, than on curves of the open road, with both superelevation and transition curves, the increase in the radius of the curve alone provides a means for increasing the speed over the curves of turnouts. Double symmetrical turnouts enable the radius of the curve to be made double that of a simple turnout, the construction being in other respects identical: they are therefore particularly suitable for high speeds while retaining safety.

CHAPTER VII.

Check rails.

Check rails, that is to say the rails by means of which on curves of the open road the guiding of the leading pair of wheels is transferred from the outer wheel to the inner wheel, are generally used mainly for reasons of safety; further they are also intended to reduce the lateral wear of the outer rail. However, as we have already explained in the preamble, check rails can only provide a limited protection against derailments. In this respect, the relation which exists between the shape of the check rail and that of the flange plays a decisive part.

In points and crossings, the check rail has the function of imparting progressively to the wheels and axles a transverse displacement, so that where the continuity of the top of the rail is broken, the flange of the outer rail is kept clear of the nose of the crossing and cannot reach the groove of the crossing. A surprising fact is that the check rail in turnouts, where in fact, the least radii of curves are to be met with, is continued over a very short length. The *normal width of the groove* is most often 41 mm. (1 39/64 inches); it is exceptionally from 44 to 45 mm. (1 47/64 to 1 49/64 inches), for check rails in turnouts and crossings of standard gauge railways. For narrow gauge tracks [1.067 m. (3 ft. 6 in.)] the Dutch East Indies State Railways use grooves 34 to 37 mm. (11/32 to 1 25/64 inches) wide; for the broad gauge tracks [1.674 m. and 1.665 m. (5 ft. 6 in. and 5 ft. 35/64 in.)] the North of Spain Railways, the Madrid-Saragossa-Alicante Railways and the Portuguese Railways use a groove width of 50 mm. (1 31/32 inches).

In curves of the open road the groove width depends upon the superelevation provided; for railways of standard gauge, it generally varies, between 41 mm. (1 39/64 in.) + gauge widening and 65 mm. (2 9/16 in.). The German State Railways have fixed, for curves in the open road of radius R = 200 m. (10 chains), the width of groove at 50 mm. (1 31/32 inches), on account of the small superelevation of these curves.

The Dutch East Indies State Railways [gauge of 1.067 m. (3 ft. 6 in.)] use a groove width of 65 mm. (2 9/16 inches) for a radius of curve R = 150 m. (7 1/2 chains) and of 60 mm. (2 23/64 inches) for R = 200 m. (10 chains).

The Portuguese Railways [gauge of 1.665 m. (5 ft. 5 35/64 in.)] adopt the following groove widths :

61 mm. (2 13/32 inches) for a radius of curve R $\geqslant$ 1 000 m. (50 chains),

76 mm. (3 inches) for a radius of curve R = 990 to 401 m. (49.5 to 20.05) chains),

86 mm. (3 25/64 inches) for a radius of curve R $\leqslant$ 400 m. (20 chains).

The *check rails in turnouts* are nearly always *superelevated* relatively to the running rail. For diamond crossings the superelevation is normally greater than for simple crossings. The superelevation varies from 7 to 50 mm. (9/32 inch to 1 31/32 inches); for diamond

crossings, in particular, it is between 30 and 50 mm. (1 3/16 and 1 31/32 inches). The German State Railways superelevate the check rail in simple crossings, 20 mm. (25/32 inch), and in diamond crossings 45 mm. (1 49/64 inches) relatively to the running rail.

The *tolerances relative to the width of the groove* are fixed in very different ways by the various railway administrations. Their limits are comprised approximately between ±2 mm. (±5/64 inch) and ± 6 mm. (± 15/64 inch). For diamond crossings in level intersections and junctions only comparatively small tolerances may be allowed, for reasons of safety. Adopting this point of view, the German State Railway has authorised for this case a tolerance of $\begin{matrix}+2\\-1\end{matrix}$ mm. $\begin{matrix}(+\ 5/64 \text{ inch})\\(-\ 5/128 \text{ inch})\end{matrix}$ only and allow for all other cases tolerances of $\begin{matrix}+5\\-2\end{matrix}$ mm $\begin{matrix}(+\ 25/128 \text{ inch}).\\(-\ 5/64 \text{ inch}).\end{matrix}$

Amongst the administrations which have replied to the questionnaire, there are only four which use *check rails in curves of the open road.* The Netherlands Railways have restricted the use of these check rails to the principal electrified lines, where they are in use for curves of radius R ⩽ 600 m. (30 chains). The German State Railways limit the use of check rails to curves of R ⩽ 300 m. (15 chains). The Danish State Railways have adopted similar practice, but also use check rails for curves of large radii on embankments. On the Dutch East Indies State Railways the use of check rails is the rule for curves of radius R = 150 m. (7 1/2 chains) and R = 200 m. (10 chains).

The opinions of the railway authorities differ on the subject of the *increase in train resistance* due to check rails. Without any doubt, check rails increase the train resistance and tend towards a reduction of speed because the inside wheel, with its point of pressure carried further forward and its inner face ordinarily not very smooth, takes up its function of guiding with greater resistance. However, the increase in the train resistance is not in any case sufficiently great for it to be noticed in practice by any decrease in the speed, especially when only short sections of track with check rails are concerned. On the other hand, the generally admitted greater safety against derailment is not judged sufficient to lead, on this assumption, to greater speeds in sections equipped with check rails. Check rails are used on down gradients, on the level and on rising gradients by the Danish State Railways, the Netherlands Railways and the Dutch East Indies State Railways. The German State Railways employ check rails only for

tracks traversed solely on the down grade,

tracks traversed principally on the down grade, if local conditions require such provision,

tracks on the level immediately continuous with fairly long sections on the down grade. Most frequently the check rails are continued through the transition curve into the straight alignment. No definite length for continuation into the straight is fixed; the Netherlands Railways alone fix this length at 6 m. (19 ft. 8 1/4 in.). The Dutch East Indies State Railways extend the check rails through the transition curve, for a curve of radius

R = 150 m. (7 1/2 chains) up to the tangent point, and for

R = 200 m. (10 chains) up to a point on the curve at a distance of 5 m. (16 ft. 5 in.) from the tangent point.

As *advantages* of the use of check rails in curves of the running road, greater security against derailment and reduced lateral wear of the outer rail are mentioned. In addition, the check rail forms a transverse bracing of the whole of the track which resists deformation

of the curves or, in other words, alteration of the curvature. As disadvantages, there are mentioned a greater train resistance and a greater initial cost of construction. But in a general way the opinion is held that the advantages outweigh the disadvantages.

Guard rails, that is to say rails fixed on the inside of the track against which the wheels only come into contact in the event of derailment, are used generally on bridges exceeding a certain length. Some of the authorities who have replied also provide them at very busy level crossings. The German State Railways in addition use guard rails at situations where it is desired to prevent derailed vehicles from dropping down or to provide bearings for scotches to prevent the vehicles moving.

The width of the groove formed by the guard rails on bridges is generally between 150 and 250 mm. (5 29/32 inches and 9 27/32 inches), more rarely 70 mm. (2 3/4 inches); it varies from 50 to 60 mm. (1 31/32 inches and 2 3/8 inches) at level crossings.

On the majority of the systems which have replied, the guard rail is fixed at the same level as the running rail; in a few instances only it is placed from 20 to 50 mm. (25/32 to 1 31/32 inches) higher.

With the exception of one railway, no case can be described in which a serious accident has been avoided through the presence of guard rails. The Netherlands Railways are unique in quoting an instance where a goods wagon which had become derailed just short of a big bridge was prevented by the guard rail from striking the beams of the bridge. As a summary, the following may be said :

Check rails can increase the safety against derailment on curves, if their profile and that of the wheel flange are of correct mutual design. They reduce the side wear of the outer rail; moreover, they brace the track as a whole transversely and thus resist alterations in curvature. Check rails in turnouts are generally higher than the running rail. Although the centre of pressure of the inner guiding wheel, located with this guiding effect on the inside face of the wheel, is transferred further forward, safety against derailment is increased, thanks to the slightly inclined surfaces of contact between the inner face of the wheel and the check rail.

Guard rails, intended to restrict as much as possible the effects of derailments, serve to improve security on bridges of a certain length or at places where it is desired to prevent the vehicles falling or colliding with an obstacle.

C. — SUMMARY.

The immunity of vehicles against derailment pre-supposes the carrying into effect of certain hypotheses relative to the construction and maintenance of the vehicles and of the track. As a result of long years of experience the essential precautions to ensure safety of the rolling stock are known. There are still certain unknown factors in the determination by calculation of the conditions to be met in order to obtain this safety, and this not only in the domain of high speeds, which question III put to the reporters ought to help to elucidate. For moderate speeds and with uniform guiding effect, the problem may be considered with reasonable approximation as being of a purely static order. In this case, it is the custom to consider the safety of running as ensured when the static guiding pressure does not exceed a given multiple of the static pressure of the guiding wheel, although it is necessary also to take account of the length of the lever arm by means of which the guiding pressure causes rotation about the « friction centre » of the vehicle or of the part of the vehicle rotated by this pressure. Now, in the case of irregularities in the track,

particularly in its alignment, and at high speeds, dynamic forces also enter into action. The guiding pressure is then equal to the sum of the « frictional guiding pressure » and of the « guiding pressure of the mass » and the vehicle no longer pivots about the friction centre, but about an axis of rotation situated generally further in the rear.

It is now easy to understand that the reporters have taken as the principal object of their questions to the railway authorities and of the use to be made of the replies received : 1. in so far as the vehicle is concerned, the method of support, the adaptation to irregular track surface, and the guiding effect; 2. in so far as the track is concerned, all which normally ensures steady support and guiding of the vehicle in a general manner. As a result of what the particulars, on which the report is based, have revealed on the matter as a whole and of what the consideration of the problem supported by calculation has taught us, we may say to sum up, either as a statement of fact or in the nature of a recommendation, the following :

a) In so far as the vehicles are concerned.

The whole of the running gear of steam locomotives, tenders, and electric locomotives and the couplings which unite the parts of the vehicle must be so arranged that turnouts and curves can be run through without restriction.

In agreement with the majority of the administrations, we recommend that the fullest use should be made of 4-wheeled bogies for guiding purposes, that the longitudinal axis of the vehicle should be guided solely by the front bogie centre and that the greatest possible guided length should be employed.

When the guiding effect is dependent upon a single pair of wheels, strong centring devices are required. The design of the under-carriages of electric locomotives and steam tank locomotives should be such that they are capable of running equally well in both directions and they should be given a long rigid wheel base, which may come into play when guiding by a single pair of wheels is insufficient.

All vehicles for fast trains should only be designed as bogie vehicles with bogies fitted with bolsters.

In the modern types of bogie having triple suspension, the once common system using transverse springs has been replaced by that in which the spring are arranged longitudinally. Advantages : long wheel base, longer springs, steadier running, better accessibility.

Double suspension bogies (American type) have bearing springs disposed transversally only.

There is an undeniable tendency towards the use of longer wheel bases than in the past, for carriage bogies. The four-wheeled bogies of the triple-suspension variety have a wheel-base of 3.60 m. (11 ft. 9 3/4 in.), the double suspension bogies of the American type have reached, in the design in use upon the Netherlands Railways, a wheelbase of 3 m. (9 ft. 10 1/8 in.).

Four and six-wheeled coaches can be used without hesitation for high speeds only if they are built with a long wheel base and if the wheels carry sufficient load. From the point of view of smooth riding, these vehicles are inferior to bogie coaches, and, were it for this reason alone, the majority of the administrations prefer not to use them for high speeds.

Buffer pressure on one side is avoided by compensating gear, and mutual oscillations of the vehicles by tight coupling.

On all sides great importance is attached to the periodical inspection of the vehicles, either at the end of a fixed term of service, or after a certain distance has been run.

On the subject of the method of sup-

port, the following is worthy of notice : out of the 30 steam locomotives mentioned, 9 (30 %) are supported on 3 points, 6 (20 %) on 4 points, 2 (6.7 %) on 5 points and 13 (43.3 %) on 6 points. 14 locomotives are supported in 2 planes and 16 in 3 planes.

Out of the 30 locomotives, 27 (90 %) have bogies with two pairs of carrying wheels and one locomotive has a Krauss-Helmholtz bogie at the leading end. 2 locomotives are fitted with pony trucks.

Among the bogies of the 28 locomotives, 10 (35.7 %) have the method of support characteristic C*e*, 3 (10.7 %) the characteristic C*me*, 6 (21.4 %) the C*s*, 4 (14.3 %) the C*m's* and 3 (10.7 %) the C*m'l*. In the case of 2 locomotives the category is not known.

None of the bogies has a transverse equaliser at the front end; the method of support characteristic C*v* is not represented amongst these steam locomotives.

Out of the 14 electric locomotives mentioned, 3 (21.4 %) are supported on 3 points, 8 (57.1 %) on 4 points, 2 (14.3 %) on 6 points, and 1 (7.2 %) is supported on 8 points. 64.2 % of the locomotives are carried in 2 planes, 21.47 % in 3 planes and 14.34 % in 4 planes.

Among these 14 electric locomotives, 3 (21.4 %) have at both ends, and 3 at one end, four-wheeled carrying bogies; of these 6 bogies, 4 (66.6 %) have the support characteristic C*v* and one (16.7 %) each the C*e* and C*me*.

The 3 locomotives fitted with bogies at one end only have pony trucks at the other end.

Among the 14 — 6 = 8 remaining locomotives, 3 (21.4 %) have bogies with one pair of carrying and one pair of driving wheels, 4 (28.6 %) have a Bissel truck at each end, 1 locomotive (7.2 %) has no pair of carrying wheels and the guiding is ensured by the six-wheeled motor bogies.

Eight-wheeled tenders should be carried on two four-wheeled bogies.

In general, the leading bogie wheels of steam locomotives are less heavily loaded than the driving and coupled wheels. The leading pony truck wheels of steam and electric locomotives are less heavily loaded to the extent of 21.8 % than the coupled wheels and axles which follow them.

The driving and coupled wheels of a steam or electric locomotive are equally loaded.

The trailing carrying wheels of steam locomotives are generally little more ligthly loaded than the coupled wheels.

In the case of all the administrations consulted, the water and coal spaces are so designed that the ratio between the wheel loads of the tender vary as little as possible relatively to one another as the quantities carried change.

For steadiness in running, the position as regards height of the centre of gravity is very important, as a high centre of gravity is favourable to smooth running. The question of safeguarding the stability on curves concerns in the first place the permanent way engineer and in the second place the operating engineer. The vehicle is necessarily constructed to meet the requirements it has to fulfil. From the point of view of sufficient static load on the leading wheels, the locomotive should be supported in two transverse planes and in consequence be carried on 3 or 4 points. In the case of locomotives, no more than 4 points of support should be employed unless the design necessitates some different arrangement. If they must be supported on 5 points this should be done by means of a spherical bogie centre.

As immunity against derailment depends so much on the bogie, it should be designed to support the main frame in two or three transverse planes and to ensure there should be no variation in load on its 4 wheels, or at the least

on the wheels of the leading axle, whilst travelling over uneven sections of track or over defective points.

It is easy to appreciate, by means of the explanations and diagrams relating to the characteristics of the bogie, which types of bogies should be used from the static point of view and what is the significance of the ratio between the flexibility of the main bearing springs and that of the bogie bearing springs. The play of the axle boxes in the axle box guides should be sufficient to ensure that the wheels and axles do not stress the frame when running over uneven sections of track. As inequalities in level of the track are inevitable, the axles must be able to take up relatively to the horizontal, an inclination of at least 1 in 60..

The considerations from which we have built up the equation

$$\frac{Y}{Q} = \frac{\sqrt{1+\frac{h^2}{l^2}}}{1+\text{cotan}\,\alpha}$$

given in Chapter IV of Part I (Vehicles) of the report, have shown that security against derailment is very great, for small striking angles, that it diminishes at first rapidly when the striking angle increases, then very little for the big striking angles, and approaches as an asymptote to the value $\frac{1}{1+\text{cotan}\,\alpha}$. On the other hand, we have also shown that the angle α of the flange has to be considered from an economic aspect and that, without the static security against derailment being affected, this angle may, in certain cases, be made less than 60°. Trials are at the present time being made with wheels with flanges at an angle of 54° only.

When considering the centring devices of guiding bogies, the great value of spring devices has been brought out and the fact that their principal object is « elastic guiding » has been shown as being indispensable if the bogies are to adapt themselves with certainty to track irregularities. Taking a modern type of bogie as example, we have shown how important it is to limit the magnitude of the centring forces in order that the guiding pressures should not assume disturbing proportions. We have explained that « bolsters » are not so efficient as spring centring gear and that helical springs are not to be recommended in the centring gear of guiding bogies.

The hope that the administrations consulted might make some useful contribution towards the numerical determination of the dynamic forces acting on the leading wheels was not realised. We have shown what were the difficulties and explained, by reference to their work, how Dauner and Uebelacker had tackled the problem. We perceive that notable progress has been made in the determination of the « reduced mass ». If special centring devices are not used, difficulties are occasioned by the value c of the « elastic flexibility » needed for determining the guiding pressure of the mass to be introduced with the frictional guiding pressure into the ratio Y : Q indicated above.

A short note on trials on the German State Railways reveals the fact that a start has been made in measuring the forces Y and Q on the vehicle *in motion*.

The locomotive ought to be capable of being guided by small turning forces (Richtkräfte). As a criterion of security against derailment we may consider the equation

$$\frac{Y}{Q} = \frac{\sqrt{1+\frac{h^2}{l^2}}}{1+\text{cotan}\,\alpha}$$

where Y is the total guiding pressure and Q the load on the leading wheel.

It is desirable that high-speed locomotives should have leading bogies with either two pairs of carrying wheels or one pair of carrying and one pair of

coupled wheels; but the latter variety should be only employed if, as in the Krauss-Helmholtz and other similar types, the tractive effort of the coupled pair of wheels does not tend to increase the striking angle of the leading pair of wheels. The axis of rotation of the bogies ought to be located between the two bogie axles.

Leading bogies of the Java type, however, formed of one pair of carrying and one pair of coupled wheels mounted in their own frame, with the axis of rotation situated to the rear of the coupled axle, have given, with about one exception, every satisfaction.

The pressure of the flanges of the leading wheels ought to be as small as possible. In consequence, the intermediate driving wheels and axles should be provided with sufficient lateral play to prevent them from exerting any lateral pressure on the locomotive frame.

Pony trucks are generally hardly suitable for guiding high-speed locomotives. When they are used, it is necessary to pay very special attention to the centring gear.

It is desirable for the guided length to be as great as possible. With a long guided length the main frame is not subjected to any stress when running through curves of large radius; moreover on curves of small radius, heavy turning and guiding forces are avoided.

High-speed locomotives ought to be fitted, at the front end, with spring centring gear. The initial and final loads on the springs ought not to differ greatly. A large amount of lateral play of the bogie may necessitate special forms of construction (cf. Winterthur type). Laminated springs are the most suitable for the usual design of centring gear. Of centring gears with bolsters, those with roller bearing parallel links relatively meet the required object best.

Modern electric long wheel base locomotives are sometimes only guided indirectly by the coupled pairs of wheels (up to 4 pairs), these wheels being given side play. In such cases centring devices with laminated springs only must be used.

Finally, we explained how, full account being taken of the importance of having the best static support, bi-lateral support of the leading bogie with the object of combating transverse vibrations due to outside causes and, thereby, ensuring safety at high speeds, had to be recommended.

A separate question is that of knowing if it is desirable, under these conditions, to give preference to the type of bogie with an equaliser at the forward end or the one with longitudinal compensating springs. Both varieties are used on high-speed locomotives.

b) So far as the track is concerned

the following may be said in resumé, either as statement of fact or as recommendation :

The relations between the speed, safety of traffic, and the track show that on the straight the speed may be increased to any desired limit whatever, provided that the track in question has adequate strength, is well adjusted in line and level and firmly braced and rests upon permanently elastic ballast with a thorougly drained formation. In circular curves, it is necessary mainly to see to the uniformity of the radius of the curve, the superelevation, and the gauge, and to provide for the joining of the straight alignment and the circular curve by a superelevation incline of regular outline, whose gradient decreases as the speed increases, because of the risk of derailment towards the outside, and coinciding as far as possible, with the transition curve. The interposition of a transition curve appreciably reduces the risk of derailment towards the inside in the event of the maximum authorised speed being exceeded. The value of the superelevation

plays a very important role in the possible increase of the speed on curves.

As regards questions of detail, our inquiry has yielded the following results :

The rails are generally designed according to different methods on the basis of the maximum wheel load, the spacing of the sleepers, and the speed. The weight of the rails in kgr. per metre, which bears a ratio varying only slightly with the moment of resistance of the rail section, depends in the first instance, despite the diversity of the methods of calculation, upon the wheel load and the spacing of the sleepers, as comparative computations have demonstrated. Long rails, fastened as rigidly as possible to sleepers on stone ballast, reduce the number of joints. The reduction in the spacing of the sleepers, in particular at the joint, by the use of the double-sleepered joint, the separate fastening of the rail of the sole plate and of the latter to the sleeper, ballast rammed or rolled, as thick as possible, on a well drained formation constantly dry, are the appropriate means to reduce twists and sags at the junction of the two running rails to such an extent that, even at high speeds, variations in the pressure on the wheels of the leading pair is kept within moderate limits.

The numerous theoretical and practical investigations published on the subject of gauge widening on curves have not resulted, up to the present, except to a limited extent, in any modifications of the regulations. However, the information which is available as to trials made with gauge increases of sometimes very small magnitude shows that attempts are being made to elucidate the effect of reducing the gauge widening.

High speed and a high standard of safety demand curves of as great a radius as possible. Attention should be paid, to a greater extent than the replies received appear to shew is the case, to the desirability of maintaining on curves, as far as possible, the maximum speed authorised for straight alignments. In many cases, this result may be obtained, without elaborate means, by increasing the radius of curves. At the same time, the economic aspect would undergo improvement. The greater the permitted superelevation, the more may the speed be increased on curves. The limit is regulated by the amount of superelevation which can be used in practice. Between two reverse curves, an intermediate straight alignment does not appear to be necessary, provided that long transition curves and gently graded superelevation inclines ensure the transit between the curve and the counter-curve. Between compound curves, it is expedient to interpose, instead of a straight, an intermediate curve of greater radius. When increasing the six-foot way of double lines at station approaches, curves of more than 10 000 m. (500 chains) radii are recommended.

With the object of maintaining the correct radius of curvature, it is helpful to use fixed reference marks.

The decisive factors for the determination of the superelevation are :

1. The safety of working at high speeds of locomotives with a high centre of gravity.

2. Steady running of the vehicles.

3. Good economic results thanks to the reduction of the lateral wear of the outer rail.

Researches made into the value of the superelevation corresponding most nearly to these conditions for standard gauge tracks have furnished proof that the economic aspect is the better safeguarded when the superelevation figure approaches the more closly to the value

$$h = 8 \frac{V^2}{R} + 30$$

while in exceptional cases it may be made equal to

$$h = 8\,\frac{V^2}{R} - 30.$$

For reasons of safety, the value of the superelevation should not fall below

$$h = 11.8\,\frac{V^2}{R} - 90.$$

In this manner, the superelevation becomes a function of the ratio $\frac{V^2}{R}$, that is to say, of the normal acceleration, whilst up to the present most of the superelevation formulæ were based upon the ratio $\frac{V}{R}$, that is to say, on the angular velocity. In calculating the superelevation, the maximum permissible speed should be adopted as the value of V. The superelevation value h may also serve as a measure of that part of the acceleration p which is neutralised out. The effective acceleration for standard track is

$$p_\Delta = \frac{V^2}{12.96\,R} - \frac{h}{153}.$$

Extended into a straight alignment, the first term of the right-hand side of the equation would disappear and at high speeds the acceleration would be reversed, and consequently the force, and this is shewn in an unfavourable manner by the unloading of the outer leading wheel. The smaller the superelevation value h becomes, the greater the effective acceleration p_Δ grows. It increases uniformly if the transition curve and the superelevation incline overlap or if, in default of any superelevation, a transition curve has been interposed. In order to obtain steady running at high speeds, it is necessary, as a general rule, that

$$p_\Delta \leqslant 0.40 \text{ m. (1.31 feet) per sec}^2,$$

whilst in exceptional cases, namely in curves of the open road and of turnouts without any superelevation,

$$p_\Delta = \text{approx. } 0.60 \text{ m. (1.97 feet) per sec}^2$$

can generally be regarded as the limiting value for reasonable and sufficiently safe running. In actual practice, it is necessary to consider the fact that p_Δ may increase materially as a result of V being exceeded and R and h being diminished. The steadiness in running and, in certain circumstances, the stability are adversely affected thereby.

The transition curve is intended to produce the progressive change in the velocity of rotation of a vehicle. On the cubic parabola, this rotation takes place with uniform acceleration. The change of acceleration

$$\Psi = \frac{p_\Delta}{t} = \frac{\left(\frac{V^2}{12.96\,R} - \frac{h}{153}\right)V}{3.6 \cdot (l + a)},$$

which depends essentially, all conditions being similar, upon the length l of the transition curve and the total wheelbase a of a vehicle, serves to measure the quality of a transition curve. The superelevation incline is usually set out in a straight line. The gradient of the superelevation incline ought to be adjusted to such a relationship with the speed that the same period of time is always available for the rotation of the vehicle. We then obtain as a figure for the gradient ratio :

$$\frac{1}{n} \leqslant \frac{1}{10\,V}.$$

If the superelevation incline is longer than the transition curve, it is necessary to prolong it into the circular curve.

The turnout curve is not so good as the curve in the open road. It is better designed, it is true, since it offers greater resistance to changes of level and of curvature, but this advantage is largely counteracted by numerous disadvantages:

absence of the transition curve, angle of intersection or incidence of the tongues, interruption of the rolling surface of the outer rail at the crossing, lowering or diminution of the superelevation at the crossing, guiding effect transferred from the outer wheel to the inner wheel at a small angle of flexion, finally multiplication of the points of shock due to short sections of rail or short crossings. The use of tangent switches with a small angle of incidence and the continuation of the junction lead across the crossing, with a lessened groove, constitutes a noteworthy improvement. A long check rail with entrance ramps inclined as little as possible can appreciably improve and render more certain the guiding of the leading pair of wheels. As in turnout curves without superelevation or transition curve, through the increased acceleration p_Δ and the change of acceleration Ψ, the speed should be less than in curves on the open road, which are superelevated and provided with transition curves, the only means of raising the speed over turnout curves is to increase the radius of the curve. Double symmetrical turnouts permit of the use of a curve of radius double that in a simple turnout, the construction being otherwise identical. They are therefore particularly suitable for high speeds, at the same time as the safety is safeguarded.

Check rails in the open road decrease the liability to derailment on curves, provided that the shape of the check rail and that of the wheel flange are designed to suit one another. They reduce the lateral wear of the outside rail, and besides, they transversely brace the track as a whole, and thereby resist any change in the curvature. Check rails in turnouts are generally raised above the running rail. Despite the transfer forward of the point of pressure of the inner leading wheel, which is located with this guiding action on the inner face of the wheel, security against derailment is enhanced owing to the slight inclination of the contact surfaces between the inner face of the wheel and the check rail.

Guard rails, intended to lessen so far as possible the effects of derailments, serve to increase safety on bridges exceeding a certain length and at situations where it is desired to prevent vehicles from dropping or coming into contact with any osbtacle.

[621. 132.8 (.62)]

Steam Rail Cars on the Egyptian State Railways,

by W. D. KNIGHT,

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PART I.

The Egyptian State Railways is one of the few Administrations which continues to favour steam for the power unit of its rail cars.

Steam was selected in preference to Diesel and petrol because the employees of the repair shops and running sheds were more familiar with the maintenance and running of the simpler mechanism involved.

The development of rail car services has however been carefully watched and information has been collected regarding the experience of railways that use other types of engines.

Up to the present the information and data, so collected, has not been sufficiently convincing to commend any alteration in the policy of using steam as the motive power for rail cars.

Diesel oil is admittedly a cheaper fuel than coal. If selection rested on the cost of fuel consumption per mile a Diesel engine would certainly prove its superiority.

Maintenance costs must however be considered. There is little doubt that an internal combustion engine, with a clutch transmission, is more expensive to maintain than a steam engine which, by reason of the expansive properties of steam, dispenses with the medium of a transmission and works at a lower speed; whilst the maintenance of a Diesel-electric engine is the most expensive of all. Therefore when maintenance is added to the cost per mile the economy effected by the use of Diesel oil is somewhat depreciated.

The most serious consideration however is the capital cost. Rail cars propelled by Diesel-electric engines cost from two to three times as much as steam cars; consequently when interest on capital and depreciation are added to the cost per mile the economy of Diesel oil disappears.

In comparing data obtained from other railways one is confronted with difficulties in selecting a basis for comparison.

Cost per car-mile conveys nothing. Some cars are much higher powered than others and some are heavier than others and the fuel consumption is increased accordingly.

The cost per horse-power-mile suggests itself as a possible basis of comparison. Against this, is the fact that cars which are apparently required for similar work vary considerably in H.P., so that, in the absence of information regarding the reason for the high powered cars, the comparison might create an incorrect impression.

Seeing, however, that the seating capacity of a car is a direct measure of its commercial value, and assuming that the H. P. of a car is a measure of its seating capacity, a basis for comparison might be sought in the cost per seat-mile. An objection to this, however, lies in the fact that cars which are designed for long-distance traffic require more comforts than cars which are intended for short runs. Also some railways are

obliged to allow a larger floor area per person than others, depending on the degree of comfort which the passengers have been educated to expect.

On the whole the cost per ton-mile is probably the fairest basis of comparison. It may be assumed that the horse-power is more or less commensurate with the weight of the car, whilst the number of passengers will not affect the gross load considerably.

Such a basis, however, does not take into consideration that the heavier built car of a given capacity will have a longer life than a lighter car of the same capacity whilst its efficiency, as measured by the infrequency of failures, will be greater.

The most disturbing factor in an international comparison is the difference in money values and relative costs of material and labour in different countries.

In some countries the cost of labour relative to materials is far higher than in other countries; certain countries can obtain fuel and materials locally whilst other countries are compelled to purchase fuel and materials in a foreign market. In addition to these disturbing factors is the complication of the rate of exchange and the question as to whether such a factor should be considered.

In a word, true comparison between the costs of running the different power units can only be effected by carrying out extensive tests on the same railway.

The foregoing remarks serve as an apology for the criticisms and comparisons that follow.

The Egyptian State Railways have for some years been exploiting eleven articulated steam rail cars of 150 H. P., weighing 52 tons, at a scheduled speed of 40 miles an hour. The cost after taking into consideration every item of expenditure including interest on capital and depreciation, works out at 10.1 pence per mile or 0.195 penny per ton-mile.

This figure is remarkably low when compared with results obtained elsewhere.

The summaries of the various reporters on Question XIII, Cairo Congress, 1933 (Use of rail motor cars on secondary lines) have been carefully scrutinized in order to ascertain the cost of running petrol-mechanical, petrol-electric and Diesel-electric cars of similar weight.

The Boston and Maine Railroad operates petrol-electric cars of 275 H. P., weighing 48 tons, and scheduled for a speed of 25 miles an hour. The total cost of running is given as 0.613 dollar per mile, which is equivalent to 0.631 penny per ton-mile.

The Chesapeake and Ohio operates a petrol-electric car of 275 H. P. weighing 59 tons, at a schedule of 30 miles per hour. The total cost of running is given as 0.412 dollar per mile which is equivalent to 0.345 penny per ton-mile.

The Great Northern Railway (U. S. A.) operates a Diesel-electric car of 400 H.P., weighing 66 tons, on a 26-mile an hour schedule. The total cost of running is 0.3061 dollar per mile, equivalent to 0.229 penny per ton-mile.

It appears therefore that the cars cited above are from 17 % to 314 % more expensive to run than the steam rail car.

Of course the higher costs of the other units are partly accounted for by the higher horse-powers.

The question therefore arises as to why, when the steam unit can haul a car of 52 tons at 40 miles an hour with 150 H.P., it should be necessary to design cars of approximately the same weight working at much lower speeds with horse-powers of 275 and even 400.

It is admitted that the steam unit does not haul a trailer whereas it is understood that the other three units do. The fact however would scarcely account for so great a difference in the horse-powers.

The Belgian National Railway Company operates a rail car which appears to be more comparable with the steam rail cars of the Egyptian State Railways.

It weighs 47 tons and is provided with a Diesel-mechanical power unit generating 150 H. P. The cost of running works out at 0.077 penny per ton-mile without allowing for interest and depreciation.

The cost of the steam rail car on the same basis of comparison works out at 0.088 penny per ton-mile.

It should be added however that the scheduled speed of the steam rail car is 40 miles per hour whilst that of the Diesel-mechanical car is 25 miles per hour — a difference which will easily account for the 14 % increase in cost of running the steam car.

The figures have been deduced without any adjustment for rate of exchange.

Whether such an adjustment should be made is open to question.

Had the Egyptian State Railways purchased the materials for running the cars from a country where the rate of exchange was unfavourable, the cost would undoubtedly have been increased. On the other hand, the railways which have been selected for comparison presumably purchased materials from other countries where the rate of exchange was favourable, and by so doing, the cost of operation was reduced.

Briefly the values in pence per mile have been deduced by converting on the old basis of exchange, viz., 486 dollars and 124 fr. to the pound sterling.

Whether or not the conversion figures are justified and whether the cost per ton-mile is a sound basis of comparison the Egyptian Railways must, in default of more convincing figures, continue to regard its steam unit as a sound financial proposition as compared with rail cars in other parts of the world.

Certain reporters have called attention to the difficulties already mentioned of comparing costs on an international basis.

The moment is therefore opportune to offer a suggestion which, though too late for the approaching Congress, may be applied on future occasions when it is desired to collect data for the purpose of comparing costs of operation.

The suggestion is as follows :

Each railway should ascertain the cost of running a standard train, i. e., a train of given load at a given speed on, as nearly as possible, a level track. The figures will of course vary considerably in different countries.

Thereafter, the cost of running a rail car should be ascertained and expressed as a fraction of the cost of running the standard train.

By this means, a figure will be obtained which will show the relative cost of running a rail car as compared with the cost of running a standard train, under identical conditions of currency, labour charges and material costs.

Such a figure admits of international comparison. Briefly, the railway which shows the lowest figure can safely claim to have the most economical car.

It has already been admitted that the cost per ton-mile is not a perfect basis of comparison because the greater the weight of a car of a given area, the longer is its life provided that the weight has been utilized wisely.

If the weight is put into the engine so that a substantial margin of horsepower is allowed over and above the amount necessary to cope with the service and assuming the engine has been designed commensurately a decrease of failures may be expected as well as longer life.

Such being the case two important items in assessing relative costs are affected, viz., interest and depreciation on capital.

The lighter car of a given area will have a shorter life and therefore its cost per mile under these headings will be correspondingly increased.

The cost per ton-mile which has been given for the steam cars of the Egyptian State Railways has been based on an assumption of a 15 years' life. The fig-

ure was arbitrarily chosen and intentionally erred on the low side. Experience with wheel rolling stock, of which the Egyptian State Railways can claim a considerable amount, indicates that the bodies and chassis will run a much longer period.

The engines, however, appear to be somewhat light for the work which is required of them. Failures have occurred which, owing to the fact that they are traceable to a variety of sources, lead to the supposition that a heavier built engine would have given more satisfactory service.

The weight of the engine without the boiler is 3 586 lb. and with the boiler 11 875 lb. which, expressed in lb. per H. P. is about 23 and 79 respectively.

Mr. Level points out in his report, that Diesel engines can be built at 16.75 lb. per H. P. (1). This is an astonishingly low figure. It would be interesting to know if any data is available regarding the life of such an engine, or if any statistics can be given regarding its failures.

It is probable that on many railways the failures of rail cars are frequently traceable to the fact that the personnel is more accustomed to the rougher handling of locomotives than the delicate mechanism of rail cars. It is also probable that such a consideration creates a certain amount of suspicion with regard to Diesel-electric cars.

There is little doubt that the more delicate and complicated mechanism involves a greater number of failures and a corresponding increase in cost of maintenance.

Apart, then, from the economical results which have been achieved with steam rail cars on the Egyptian State Railways, the fact should not be overlooked, that the results have been obtained by a simple design and a long-established prime mover.

(1) *Bulletin of the Railway Congress*, June 1932, p. 1139.

PART II

The relative merits of a rail car as compared with a road car is not only an intriguing subject for consideration, but it is a most important point that must be settled in the near future.

There is no doubt that a rail car appears to possess some outstanding advantages.

For example, the cost of running, excluding interest and depreciation on capital and without considering the costs of track or road maintenance or any charges other than those appertaining to the Mechanical Department, is entirely favourable to the rail car. A rail car can carry 100 people at 4.6 pence per mile whilst a road car is capable of carrying 25 people at a cost of not less than 2.5 pence per mile.

Moreover, a rail car is for a given weight and horse-power capable of higher speeds than a road car.

The question therefore arises as to why rail cars are not recovering the traffic which road cars have secured.

The answer to the question may be solved by considering the outstanding disadvantages to which a rail car is subjected.

On the single railway line, two rail cars cannot pass one another. Special arrangements must therefore be established to insure that two cars are not running in opposite directions within defined limits at the same time.

Road cars are not hampered by such restrictions. There are few roads, if any, in the world, where two cars cannot pass one another.

If a rail car breaks down on the single line it entirely obstructs all other cars in both directions until relief arrives, which, incidentally, may result in many hours detention.

On the other hand, if a road car breaks down it can be pushed to the side of the road in a few minutes, leaving a free

passage for approaching or following cars.

On a double railway line the conditions are not much better.

A rail car that breaks down must remain where it is, until relief arrives, obstructing all following traffic.

Whatever speed it may be capable of, it cannot pass a preceding car or train on account of the restriction of the track, and should it come upon such a car or train uncautiously a collision is inevitable.

If, on the other hand, a road car suddenly overtakes a preceding car it can swerve to one side and thus avoid a collision.

Although a rail car is capable of higher speeds, its advantage in this respect can only be exploited by the expensive introduction of signals and cabins, working on the absolute block system.

If such precautions were eliminated, the rail car could not be run as quickly as the road car with the same degree of safety.

To test the accuracy of this statement one has only to ask oneself whether one would be as content to travel in a rails-car at, say, thirty miles per hour on an unprotected railway with unscheduled traffic approaching and following, as one would be to motor at the same speed on an unfamiliar road.

Briefly, the railway track is an encumbrance rather than the potential asset which some members have described it.

There is also the question of unscheduled stops to be considered.

It is not clear why a road car should have an advantage in this respect.

It would appear that railway officials have a fixed idea that their conveyances must only stop at certain predetermined points, where elaborate preparations in the form of buildings, staff, platforms and signals must be provided; and that the time of arrival and departure must be correct to seconds and that the period of stoppage must be rigidly adhered to even though there are no passengers who wish to entrain or detrain.

With the same inconsequence they appear to be under the impression that intervening points where passengers may wish to entrain or detrain must be ignored unless they are provided with platforms, signals, etc.

Moreover, railwaymen seem to consider themselves under an obligation to find seating accommodation for every passenger who is waiting on the platform to travel. Their attitude in this respect results in the uneconomical policy of supplying trains or cars of greater capacity than normal requirements justify.

There is a fixed idea amongst railwaymen that the tariff for travelling, assessed on a mileage basis, must be universal throughout the railway, regardless of the fact that the cost of running a rail car is considerably less than running a train.

Until such ideas are uprooted from their minds of railwaymen, there is little hope for the rail car to compete successfully against the road car.

Probably the only condition under which rail cars can compete successfully against road cars is in a sparsely populated country where towns are few and far between.

Under such conditions the disadvantages of rail cars are not so pronounced.

The towns and villages in such an area being far apart admit of long non-stop runs, whilst signal cabins installed at the stations guarantee the safety of the line and admit of speeds far in excess of those that are considered safe for road cars to travel at; incidentally the dearth of traffic permits of the infrequent service that is necessitated by absolute block control.

Such a service should provide for optional stops between towns at points where passengers might wish to entrain or detrain, but no personnel should be provided at such points.

Tickets should be sold on the cars so as to insure that the company is not

under an obligation to carry anybody who had not secured a place in the car.

It is very unlikely that rail cars will ever operate successfully in densely populated areas. A frequent service involves short block sections, rapid acceleration and deceleration, also facilities for increasing the capacity of the cars or trains without retaining in continual service a hauling unit which is normally more powerful than the service demands.

The more populated is the area to be served, the less adaptable is the rail car, because whilst designers, endeavouring to adapt a rail car service to meet the exacting conditions, will find themselves led into the only possible solution, viz., electrification.

CORRIGENDUM.

Bulletin, June 1932.

Report by Herr STUDENT, Question IV, Caïro Congress.

Page 991, 1st column, 2 last lines and 4 lines at the top of the 2nd column.

Instead of :

« The Finnish State Railways... the date arrives. »

Read :

The Finnish State Railawys only carry out the boiler examinations at most *6 months* before the legal date.

Page 1022, 1st column, 23rd to 32nd lines.

Instead of :

« 5. The answer reads... must be made. »

Read :

As a rule the boiler examinations are carried out at most 6 months before the legal date, when other repairs are found to be necessary, thereby ensuring as far as possible the economical use of the locomotive.

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PUBLISHED UNDER THE SUPERVISION OF

P. GHILAIN,

General Secretary of the Permanent Commission of the International Railway Congress Association.

(JULY 1932)

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Die **Umformerlokomotive** System Oc. 8. S. W. 1082.001 der Österreichischen Bundesbahnen. (1 400 Wörter & Abb.)

1932 621 .132.8
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LÜBSEN (W.). — **Turbinen-Lokomotiven Bauart « Ljungström ».** (1 800 Wörter & Abb.).

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1932 656 .223.2 (.43)
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SCHROEDER. — Die Verwendung von **Grossgüterwagen** im Verkehr der Reichsbahn. (3 Seiten.)

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VOGT. — Wie unterstützt die Reichsbahn das Bemühen des Reichs, **die Preise zu senken und der Wirtschaft zu helfen ?** (7 1/2 Seiten & Diagr.)

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KISSLING. — Die **Wirtschaftlichkeit der Bahnsteigsperre** bei der Deutschen Reichsbahn. (4 Seiten.)

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ALEXANDER (J.). — Betrieb einer steilen **Reibungsbahn** in Guatemala. (3 200 Wörter & Abb.)

1932 621 .43 (.494)
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ÖFVERHOLM (I.). — **Elektrische Wagenheizung** mit selbsttätiger Umschaltung für 3 000, 1 500 oder 1 000 Volt. (500 Wörter & Abb.)

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1932 625 .246
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1932 **625 .143. (0**

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BLOCH (A.). — Die **Stabilität des lückenlosen Gleises.** (5 200 Wörter & Abb.)

1932 **625 .143. (0**

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WATTMANN. — Knicksicherheit von Gleisen. (1 500 Wörter & Abb.)

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PETERSEN (E.). — Ist das Doppel-Rüpingverfahren für die **Tränkung von Buchenholz** genügend ? (900 Wörter & Abb.)

1932 **656 .212.5 & 625 .258**

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GARBERS. — Die Profilgestaltung der **Ablaufberge** unter besonderer Berücksichtigung der Bremsanordnung. (10 000 Wörter & Abb.)

1932 **625 .13 (.432)**

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TRAMM. — **Verstärkung und Umbau von Schmalspurbrücken** im Bezirke der Reichsbahndirektion Dresden. (900 Wörter & Abb.)

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1932 **656 .222.1**

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MECKEL (A.). — **Lokomotivbremsung** bei hohe Geschwindigkeiten. (1 200 Wörter & Abb.)

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)932 656 .234 (.43)
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Boilers at Helsingfors power station. (3 500 words & fig.)

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The manufacture of **staybrite steel sheets and strip.** (2 700 words & fig.)

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TROEMEL (I.). — New York **subway construction. VIII. Underpinning a subway** crossing. (3 200 words & fig.)

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Missouri-Kansas-Texas constructs longest lift span. (3 900 words & fig.)

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VAN NESS (R. A.). — **Treated timber on the Santa Fe after fifty-seven years.** (1 400 words & fig.)

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Avonside Diesel locomotives. (1 500 words & fig.)

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Harding Rhodes **gear change indicator.** (500 words fig.)

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1932 **656**

Revue générale des chemins de fer, juin, p. 509.

LE BESNERAIS (R.). — La **sécurité** des voyageurs en chemin de fer. (18 200 mots & fig.)

1932 **621 .43 (.4**

Revue générale des chemins de fer, juin, p. 564.

Résultats d'essais de la **locomotive Diesel à air comprimé** des Chemins de fer du Reich. (1 900 mots & fig.)

Revue politique et parlementaire. (Paris.)

1932 **385. (09 (.4**

Revue politique et parlementaire, 10 juin, p. 542.

VIATOR. — Les **chemins de fer** en Russie soviétique au seuil de 1932. (2 900 mots.)

Revue universelle des Mines. (Liége.)

1932 **624 .5 (.7**

Revue universelle des mines, n° 11, 1er juin, p. 48

n° 12, 15 juin, p. 514.

TOURNAY (Ch.). — Le **pont** George Washington New-York. (4 300 mots & fig.) (A suivre.)

In German.

Archiv für Eisenbahnwesen. (Berlin.)

1932 **656 .21**

Archiv für Eisenbahnwesen, März-April, S. 297.

GIESE (K.). — Der **Hafenbahnvertrag** in Theorie und Praxis. Zur Frage der Organisation des Hafenbahnbetriebs. (17 900 Wörter.)

1932 **656 (.4**

Archiv für Eisenbahnwesen, März-April, S. 345.

GRETSCH (Dr. R.). — Die **einheitliche Verkehrsregelung** in England. (10 000 Wörter & 2 Tafeln.) (Fortsetzung.)

1932 **385 .1 (.7**

Archiv für Eisenbahnwesen, März-April, S. 381.

MERKERT (E.). — Die **gefährdete Lage** der amerikanischen Eisenbahnen. (11 000 Wörter, 2 Tafeln Abb.) (Schluss.)

1932 **313 .385 (.489**

Archiv für Eisenbahnwesen, März-April, S. 442.

Die **Eisenbahnen in Dänemark** in den Betriebsjahren 1928-1929 und 1929-1930. (1 500 Wörter.)

1932 **313 .385 (.485**

Archiv für Eisenbahnwesen, März-April, S. 454.

THOMSON. — Das **schwedische Eisenbahnnetz** 192 und 1929. (2 000 Wörter.)

1932 313 .385 (.481)
Archiv für Eisenbahnwesen, März, S. 469.
THOMSEN. — Die **Eisenbahnen in Norwegen** in den Jahren 1928-29 und 1929-30. (2 200 Wörter.)

1932 313 .385 (.439)
Archiv für Eisenbahnwesen, März, S. 481.
ROMAK (B.). — Die **Königlich-Ungarischen Staatsbahnen** im Betriebsjahr 1928-29. (4 000 Wörter.)

1932 385. (09 (.495)
Archiv für Eisenbahnwesen, März, S. 501.
Die **Eisenbahnen Griechenlands.** (500 Wörter.)

Die Lokomotive. (Wien.)

1932 621 .13 (.42)
Die Lokomotive, Juni, S. 97.
STEFFAN. — Die Ziele im Bau englischer **Schnellzuglokomotiven.** (10 500 Wörter & Abb.)

1932 621 .131.1
Die Lokomotive, Juni, S. 112.
Die Grenzen der **Dampflokomotive** hinsichtlich Zugkraft und Geschwindigkeit. (2 600 Wörter.)

Elektrische Bahnen. (Berlin.)

1932 621 .335 (.494)
Elektrische Bahnen, Mai, S. 101.
STEINER (F.). — Über einen neuen **Einphasenmotorwagen** der Maschinenfabrik Oerlikon (M. F. O.) für die Schweizerischen Bundesbahnen (S. B. B.). (4 400 Wörter & Abb.)

1932 625 .216
Elektrische Bahnen, Mai, S. 108.
BALKE (H.). — **Selbsttätige Steuerstrom-Kupplung** mit Starkstromkontakten für Mittelpufferkupplung elektrischer Bahnen. (1 600 Wörter & Abb.)

1932 621 .33
Elektrische Bahnen, Mai, S. 111.
FORWALD (H.). — **Streckenimpedanz und Schienenstrom** bei Einphasenbahnen. (4 700 Wörter & Abb.) (Schluss.)

Glasers Annalen. (Berlin.)

1932 621 .43 (.43)
Glasers Annalen. Heft 10, 15. Mai, S. 85; Heft 11, 1. Juni, S. 97.
NIEDERSTRASSER (L.). — Bauliche Entwicklung der **Kleinlokomotiven** der Deutschen Reichsbahn. (8 500 Wörter & Abb.) (Schluss folgt.)

1932 621 .8
Glasers Annalen, Heft 11, 1. Juni, S. 90.
LEHMANN (E.). — Ein neuer genormter **Hängekran.** (1 400 Wörter & Abb.)

Organ für die Fortschritte des Eisenbahnwesens. (Berlin.)

1932 621 .132.6 (.436)
Organ für die Fortschritte des Eisenbahnwesens, Heft 11, 1. Juni, S. 218.
LEHNER. — **2-C-2 Heissdampf-Zwillings-Tenderlokomotive** Reihe 729 der Österreichischen Bundesbahnen. (5 200 Wörter & Abb.)

1932 625 .143.5
Organ für die Fortschritte des Eisenbahnwesens, Heft 12, 15. Juni, S. 225.
SALLER (H.). — **Blattfederoberbau** nach Rüping. (4 700 Wörter & Abb.)

1932 625 .253
Organ für die Fortschritte des Eisenbahnwesens, Heft 12, 15. Juni, S. 231.
HILDEBRAND (W.). — Entwicklungsgedanken der **Hildebrand-Knorr-Bremse.** (2 800 Wörter & Abb.)

Zeitschrift des Vereines Deutscher Ingenieure. (Berlin.)

1932 62. (01 & 621 .116
Zeitschrift des Vereins deutscher Ingenieure, Nr. 23, 4. Juni, S. 551.
DAEVES (K.), PFEIFFER (E.) & URBANCZYK (G.). — **Werkstoffuntersuchungen** an alten Dampfkesseln. (3 900 Wörter & Abb.)

1932 621 .134.3 (.44)
Zeitschrift des Vereins deutscher Ingenieure, Nr. 23, 4. Juni, S. 561.
ROTHHAAS (L.). — **60 at-Hochdrucklokomotive** der Paris-Lyon-Mittelmeer-Bahn nach dem Schmidtschen Hochdruckverfahren. (1 400 Wörter & Abb.)

Zeitung des Vereins deutscher Eisenbahnverwaltungen. (Berlin.)

1932 385
Zeitung des Vereins deutsch. Eisenbahnverw., Nr. 22, 2. Juni, S. 479.
von BECK (E.). — Beiträge zur **Anpassung des Verkehrswesens an die Wirtschaftslage.** (7 500 Wörter.)

1932 656 .224 (.43)
Zeitung des Vereins deutsch. Eisenbahnverw., Nr. 23, 9. Juni, S. 497.
BAUMGARTEN. — Der **Personenverkehr** der Deutschen Reichsbahn im Jahre 1931. (7 900 Wörter.)

1932 385 .1 (.492)
Zeitung des Vereins deutsch. Eisenbahnverw., Nr. 24, 16. Juni, S. 517.
ELIAS (P.). — Die **Eisenbahnen in den Niederlanden.** (2 800 Wörter & Abb.)

1932 **625 .2 (01**
Zeitung des Vereins deutsch. Eisenbahnverw., N. 24, 16. Juni, S. 522.
WENTZEL. — Untersuchungsanlage für das Studium des **Rollvorgangs und des Laufwiderstands** von Schienenfahrzeugen. (2 000 Wörter & Abb.)

1932 **656 .212.5**
Zeitung des Vereins deutsch. Eisenbahnverw., Nr. 25, 23. Juni, S. 542.
WERNER. — Das **Räumen der Richtungsgleise** in Verschiebebahnhöfen. (1 100 Wörter & Abb.)

In English.

Engineer. (London.)

1932 **624 .32 (.66)**
Engineer, No. 3986, June 3, p. 603.
The Benue **bridge** in Nigeria. (1 900 words & fig.)

1932 **385. (072**
Engineer, No. 3986, June 3, p. 609.
New Canadian **research laboratories.** (3 600 words & fig.)

1932 **385. (072 (.42)**
Engineer, No. 3986, June 3, p. 611.
Research on the L. M. S. Railway. (3 000 words.)

1932 **621 .33 (.44)**
Engineer, No. 3986, June 3, p. 612.
Electrification of French Railways. (2 800 words.)

1932 **625 .144.4 (.42), 625 .172 (.42) & 625 .46 (.42)**
Engineer, No. 3986, June 3, p. 622.
L. M. S. Railway **plough brake van.** (450 words.)

1932 **656 .222.1 (.42)**
Engineer, No. 3987, June 10, p. 638.
BAKER (H.). — The Great Western record. (1 600 words.)

1932 **62. (01**
Engineer, No. 3987, June 10, p. 644.
Self-indicating universal testing machine. (2 300 words.)

1932 **62. (01 & 621 .13 (0**
Engineer, No. 3988, June 17, p. 656.
REED (B.). — **Pressure and stresses in locomotive parts.** (5 300 words, tables & fig.)

1932 **669 .1**
Engineer, No. 3988, June 17, p. 663.
Stainless iron and steel production. (3 400 words.)

1932 **625 .13 (.45)**
Engineer, No. 3988, June 17, p. 671.
An interesting **bridge replacement** operation. (1 500 words & fig.)

1932 **625 .246 (.4**
Engineer, No. 3988, June 17, p. 674.
An **oil-electric shunting locomotive.** (600 words & fig

1932 **625 .246 (.4**
Engineer, No. 3988, June 17, p. 675.
G. W. R. **tracks for milk tank trailers.** (300 wor & fig.)

Engineering. (London.)

1932 **62. (01 & 6**
Engineering, No. 3464, June 3, p. 645.
DOCHERTY (J. G.). — **Bending tests** on geomet cally **similar notched bar specimens.** (4 700 words fig.)

1932 **625 .4 (.7**
Engineering, No. 3464, June 3, p. 647.
SKINNER (F. W.). — New York **electric subw construction methods.** VIII. — (3 000 words & fig.)

1932 **614**
Engineering, No. 3464, June 3, p. 650.
Fire protection of oil and petrol storage tanks. (9 words & fig.)

1932 **621 .134**
Engineering, No. 3464, June 3, p. 667.
OUDET (J.). — Recent developments of the fo **cylinder compound locomotive.** (1 400 words & fig.)

1932 **621 .8**
Engineering, No. 3464, June 3, p. 670.
BOSWALL (R. O.). — The film **lubrication** of t **journal bearing.** (4 300 words, tables & fig.)

1932 **656. (06 (.4**
Engineering, No. 3465, June 10, p. 682.
Railway rationalisation. (4 200 words & fig.)

1932 **621 .18 (.89) & 621 .31 (.8**
Engineering, No. 3466, June 17, p. 703.
Fuel consumption tests : State **electric power st tion** Montevideo. (4 400 words & fig.)

1932 **656. (06 (.4**
Engineering, No. 3466, June 17, p. 719.
Fuel research and transport. (3 100 words.)

Engineering News-Record. (New York.)

1932 **625 .13 (.7**
Engineering News-Record, No. 20, May 19, p. 716.
RUSSELL (T. F.). — Bottom heading driving on C nadian Pacific Railway **tunnel** in Quebec. (1 600 wor & fig.)

1932 **624 .8 (.7**
Engineering News-Record, No. 21, May 26, p. 755.
Longest **railway lift bridge** is built by Missour Kansas-Texas Railroad. (1 500 words & fig.)

1932 **625 .13 (.73)**
ngineering News-Record, No. 21, May 26, p. 757.
EXUM (J. P.). — **Revamping a 28-year old steel-uss bridge.** (1 500 words & fig.)

1932 **62. (01 & 721 .3**
ngineering News-Record, No. 21, May 26, p. 764.
Tests provide new data on **concrete-column design.** 400 words, 3 tables & fig.)

1932 **621 .392 & 721 .9**
ngineering News-Record, No. 21, May 26, p. 769.
SEELYE (E. E.). — Assuming responsibility for **ructural welding. Precautions and procedure.** (2 200 ords & fig.)

1932 **721 .4 (.73) & 725 .31 (.73)**
ngineering News-Record, No. 23, June 9, p. 817.
PAYZANT (O. S.). — Parallel trusses carry **dome of ncinnati station.** (2 300 words & fig.)

1932 **624. (0 & 624 .51**
ngineering News-Record, No. 23, June 9, p. 828.
BEGGS (G. E.), TIMBY (E. K.) & BIRDSALL (B.). **Suspension bridge stresses** determined by model. 000 words & fig.)

Great Western Railway Magazine. (London.)

1932 **625 .172 (.42) & 625 .175 (.42)**
reat Western Ry. Magazine, June, p. 219.
CROOK (F. C.). — The « economic » and « motor nomic » systems of **track maintenance.** (1 300 words fig.)

Indian Railway Gazette. (Calcutta.)

1932 **621 .33 (.494)**
dian Ry. Gazette, May, p. 115.
The Swiss Federal Railways. **Electric traction on the tthard line.** (2 400 words & fig.)

Journal, Institution of Engineers, Australia. (Sydney, N. S. W.)

1932 **621 .392 (.944) & 624. (.944)**
urnal, Institut. of Engineers, Australia, April, p. 113.
COCKBURN (G. R.). — **Electric arc welding as pplied to railway bridges,** with particular reference the railway bridge over the Hunter River at Single-n, New South Wales. (5 900 words & fig.)

Journal, Institute of Transport. (London.)

1932 **621 .33 (.54)**
urnal, Institute of Transport, June, p. 411.
LYDALL (F.). — **The electrification of the subur-an and certain main-line sections** of the Great Indian eninsula Railway. (17 000 words.)

1932 **388**
Journal, Institute of Transport, June, p. 440.
PEPLER (G. L.). — **Town planning and transport.** (12 000 words & fig.)

Journal, Permanent Way Institution. (London.)

1932 **625 .14 (01**
Journal, Perm. Way Institution, April, p. 70.
WALKER (R. D.). — Modern trends in **permanent way maintenance and design.** (14 400 words & fig.)

1932 **625 .143.2 & 625 .15**
Journal, Perm. Way Institution, April, p. 104.
HILL (T.). — **Manganese steel switches, crossings, and rails.** (3 800 words & fig.)

1932 **625 .142.2 (.66)**
Journal, Perm. Way Institution, April, p. 113.
WALMSLEY (H.). — **Wood sleepers** in Sudan. (5 300 words.)

Locomotive, Railway Carriage and Wagon Review (London)

1932 **621 .131.2**
Loc., Ry. Carr. & Wag. Review, June 15, p. 209.
PHILLIPSON (E. A.). — **Steam locomotive design :** data and formulæ. (1 400 words & fig.)

1932 **621 .335 (.47) & 621 .43 (.47)**
Loc., Ry. Carr. & Wag . Review, June 15, p. 211.
The **Diesel-electric locomotive** in Russia. (3 800 words & fig.)

1932 **621 .335**
Loc., Ry. Carr. & Wag. Review, June 15, p. 216.
Electric locomotive design : V. (1 200 words & fig.)

Modern Transport. (London.)

1932 **656 .212.6 (.42) & 656 .213 (.42)**
Modern Transport, No. 688, May 21, p. 3.
BINNS (A.). — **Mechanical equipment of the Port of London.** (4 200 words & fig.)

1932 **625 .23 (0**
Modern Transport, No. 688, May 21, p. 7.
Can **train weights** be reduced ? —Possibilities in use of **omnibus bodies.** (1 000 words & 1 table.)

1932 **621 .138.5 (.42)**
Moden Transport, No. 689, May 28, p. 4.
WOOD (W. V.). — Development of **costing methods** on L. M. S. R. (4 300 words & fig.) (To be continued.)

1932 **625 .26 (.42)**
Modern Transport, No. 689, May 28, p. 4.
Section, p. III.
Labour-saving methods in carriage and wagon build-ing at Derby Works, L. M. S. R. (5 500 words & fig.)

1932 **656 .253 (.42)**
Modern Transport, No. 689, May 28, Buxton Congress Section, p. VIII.
Four-aspect installation at Victoria and Exchange stations, Manchester, L. M. S. R. (2 800 words & fig.)

1932 **621 .33 (.42)**
Modern Transport, No. 689, May 28, Buxton Congress Section, p. XIV.
Electrification of Manchester, South Junction and Altrincham Railway. **Conversion on 1 500-volt direct current system,** as recommended by Railway Electrification Committee of 1927. (4 300 words & fig.)

1932 **656 .233 (.42)**
Modern Transport, No. 690, June 4, p. 3.
Pooling of traffic. (3 200 words.)

1932 **621 .132.3 (.41) & 656 .222.1 (.41)**
Modern Transport, No. 690, June 4, p. 5.
Compound locomotives for Ireland. (1 200 words & fig.)

1932 **621 .138.5 (.42)**
Modern Transport, No. 690, June 4, p. 6.
WOOD (W. V.). — Development of **costing methods** on L. M. S. R. (4 300 words & fig.) (Concluded.)

1932 **656 .213 (.42)**
Modern Transport, No. 690, June 4, Buxton Congress Section, p. V.
The Manchester **ship canal and docks.** (4 200 words & fig.)

1932 **621 .43 (.494)**
Modern Transport, No. 690, June 4, Buxton Congress Section, p. IX.
Diesel engines for rail transport. (2 100 words & fig.)

1932 **656 .212.6 (.42) & 656 .213 (.42)**
Modern Transport, No. 690, June 4, Buxton Congress Section, p. XV.
Handling at docks and terminals. (2 800 words & fig.)

1932 **656 .212.6 (.42)**
Modern Transport, No. 690, June 4, Buxton Congress Section, p. XXI.
Modern **mechanical equipment at port of Manchester warehouses.** (1 300 words & fig.)

1932 **656 .212.9 (.42)**
Modern Transport, No. 690, June 4, Buxton Congress Section, p. XXII.
Problem of internal transport. — **Movement of goods in factories and works.** (1 800 words & fig.)

1932 **656. (06 (.42)**
Modern Transport, No. 691, June 11, p. 3.
The **Buxton Congress, Institute of Transport.** (2 400 words.)

1932 **625 .26 (.42)**
Modern Transport, No. 691, June 11, p. 5.
LEMON (E. J. H.). — **Rationalisation** on the L. M. S. R. (11 400 words.)

1932 **621 .13**
Modern Transport, No. 691, June 11, p. 10.
WHEELER (R. V.). — **Fuel research in relation transport.** (6 800 words.)

1932 **385. (072 (.4**
Modern Transport, No. 691, June 11, p. 15.
HARTLEY (Sir Harold). — **Scientific research** the London Midland & Scottish Ry. (8 600 words.)

1932 **656 .1 (.42) & 656 .261 (.4**
Modern Transport, No. 692, June 18, p. 7.
Milk transport on the Great Western Railway. (7 words & fig.)

Proceedings, American Society of Civil Engineer (New York.)

1932 **532. (**
Proc., Amer. Soc. of Civil Eng., May, p. 729.
Standard symbols and glossary for hydraulics a irrigation compiled by the special committee on ir gation hydraulics. (11 600 words.)

1932
Proc., Amer. Soc. of Civil Eng., May, p. 775.
PAASWELL (G.), GRIFFITH (J. H.) & GRUN (H. E.). — **Soil mechanics research.** (5 800 words & fi

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Proceedings, Institution of Mechanical Engineer (London.)

1931 **625 .142**
Proc., Instit. of Mech. Engineers, vol. 121, p. 315.
CARPMAEL (R.). — The manufacture and use **steel railway sleepers.** (28 000 words & fig.)

1932
Proc., Institut. of Mech. Engineers, vol. 121, p. 465.
DYMOND (A. W. J.). — Some factors affecting t **riding of coaching stock.** (25 000 words & fig.)

1932 **621 .85, 621 .86 & 621 .8**
Proc., Institut. of Mech. Engineers, vol. 121, p. 559.
BEEBY (W. L.). — The **handling of materials in mass-production factory.** (4 200 words & fig.)

1932 **621**
Proc., Institut. of Mech. Engineers, vol. 121, p. 573.
RUSSEL (C. H.). — Modern **machine-tools.** (4 2 words & fig.)

Railway Age. (New York.)

1932 **625 .18 (.7**
Railway Age, No. 22, May 28, p. 896.
Norfolk & Western cleans up roadway stock. (1 7 words & fig.)

1932 **385 .581 (.7**
Railway Age, No. 22, May 28, p. 898.
Six-hour day investigation. (3 300 words.)

1932 **625 .142 (.73)**
Railway Age, No. 22, May 28, p. 901.
Railways face shortage of **ties.** (3 200 words.)

1932 **621 .134.1, 621 .135.2 & 625 .214**
Railway Age, No. 22, May 28, p. 905.
Railroad lubrication discussed at A. S. M. E. conference. (2 600 words.)

1932 **656 .255 (.73)**
Railway Age, No. 22, May 28, p. 907.
Centralized traffic control on the Boston & Maine. (400 words & fig.)

1932 **656 .225 (.73) & 656 .261 (.73)**
Railway Age, No. 22, May 28, p. 916.
Another North Shore Line innovation. **Loaded (motor) trucks now handled on flat cars.** (1 900 words & fig.)

1932 **656 .21 (.73) & 656 .22 (.73)**
Railway Age, No. 23, June 4, p. 936.
TEAL (J. E.). — How the Chesapeake & Ohio has increased **its operating efficiency.** (3 400 words & fig.)

1932 **614 .7 (.73) & 621 .133.4 (.73)**
Railway Age, No. 23, June 4, p. 941.
Fighting smoke in Hudson county, New Jersey. (1 800 words & fig.)

1932 **385 .113 (.42)**
Railway Age, No. 23, June 4, p. 952.
COCK (C. S.). — British railways show fight. (3 800 words & fig.)

1932 **621 .133.7 (.73) & 725 .33 (.73)**
Railway Age, No. 23, June 4, p. 973.
Better water conditions pay large dividends. (2 700 words & fig.)

1932 **625 .214 (.73)**
Railway Age, No. 24, June 11, p. 976.
Journal packing reclamation plant saves 82 per cent. (200 words, tables & fig.)

1932 **385 .581 (.73)**
Railway Age, No. 24, June 11, p. 979.
Test of **six-hour day proposed.** (2 300 words.)

1932 **621 .335 (.73)**
Railway Age, No. 24, June 11, p. 983.
Design of new **electric locomotives** substantiated by performance. (2 200 words & fig.)

Railway Engineer. (London.)

1932 **656 .254 (.73)**
Railway Engineer, June, p. 202.
Automatic train control in America. (6 500 words.)

1932 **621 .132.3 (.42)**
Railway Engineer, June, p. 202.
A great locomotive. (900 words.)

1932 **621 .132.8 (.82)**
Railway Engineer, June, p. 203.
Locomotives for waterless countries. (400 words.)

1932 **621 .132.7 & 621 .43**
Railway Engineer, June, p. 204.
GRIME (T.). — **Steam or Diesel for shunting?** (4 300 words & fig.)

1932 **656 .254 (.42)**
Railway Engineer, June, p. 208.
Improvements in the **working of the Lickey incline,** L. M. S. R. (1 700 words & fig.)

1932 **621 .33 (.436)**
Railway Engineer, June, p. 211.
Main-line electrification in Austria. (3 400 words & fig.)

1932 **621 .392 (.42) & 624 (.42)**
Railway Engineer, June, p. 215.
Welding of railway bridges on the London & North Eastern Ry. (2 200 words & fig.)

1932 **621 .9 (.42)**
Railway Engineer, June, p. 221.
New **machine tools** for railway workshops. (1 200 words & fig.)

1932 **621 .94 (.42)**
Railway Engineer, June, p. 223.
A new **grinding machine.** (1 000 words & fig.)

1932 **621 .132.3 (.44)**
Railway Engineer, June, p. 225.
A noteworthy **French locomotive** development. (3 000 words & fig.)

1932 **621 .334 (.42)**
Railway Engineer, June, p. 228.
Mobile trucks for railway use. — VI. The Slingsby-**automatic battery trucks.** (1 600 words & fig.)

1932 **621 .132.8 (.82)**
Railway Engineer, June, p. 230.
A new **condensing locomotive.** (4 500 words & fig.)

Railway Engineering and Maintenance. (Chicago.)

1932 **621 .133.7 (.73) & 725 .33 (.73)**
Railway Engineering & Maintenance, June, p. 383.
The Baltimore & Ohio is « sold » on **water treatment.** (4 700 words & fig.)

1932 **385 .586 (.73) & 625 .17 (.73)**
Railway Engineering & Maintenance, June, p. 387.
VESS (H. B.). — **Training foremen for maintenance work.** (4 700 words & fig.)

1932 **625 .123 (.73)**
Railway Engineering & Maintenance, June, p. 391.
Stopping slides with an effective **drainage system.** (1 500 words & fig.)

Railway Gazette. (London.)

1932 **656 .257 (.54)**
Railway Gazette, No. 21, May 20, p. 743.
Power signalling installation in India. (2 300 words & fig.)

1932 **621 .43**
Railway Gazette, No. 21, May 20, p. 748.
MIALL (S.). — **Transmissions for Diesel locomotives and railcars.** (800 words & fig.)

1932 **621 .43 (.44)**
Railway Gazette, No. 21, May 20, p. 751.
Diesel railcars for service in France. (1 300 words & fig.)

1932 **621 .335 (.593) & 621 .43 (.593)**
Railway Gazette, No. 21, May 20, p. 752.
Diesel-electric stock in Siam. (1 000 words & fig.)

1932 **621 .43 (.42)**
Railway Gazette, No. 21, May 20, p. 754.
Dorman-Ricardo Diesel engine. (600 words & fig.)

1932 **656 .1 (.54)**
Railway Gazette, No. 22, May 27, p. 773.
Road and railway competition in India. (1 100 words.)

1932 **625 .172 (.42)**
Railway Gazette, No. 22, May 27, p. 774.
An **automatic recording machine for track defects.** (1 900 words & fig.)

1932 **625 .26 (.42)**
Railway Gazette, No. 22, May 27, p. 779.
Mass production of wagons at Derby, L. M. S. R. (1 200 words & fig.

1932 **621 .43 (.42)**
Railway Gazette, No. 22, May 27, p. 782.
Drewry **Diesel-engined shunting locomotive.** (600 words & fig.)

1932 **621 .135.2**
Railway Gazette, No. 22, May 27, p. 784.
An important development in **ball and roller bearings.** (1 000 words & fig.)

1932 **656 .233 (.42)**
Railway Gazette, No. 23, June 3, p. 805.
L. M. S. and L. N. E. R. **pooling scheme.** (2 900 words.)

1932 **625 .26 (.42)**
Railway Gazette, No. 23, June 3, p. 807.
LEMON (E. J. H.). — **Rationalisation** on the L. M. S. R. — A description of the steps taken to effect economies and increase efficiency. Results achieved. (2 700 words.)

1932 **385. (072 (.42)**
Railway Gazette, No. 23, June 3, p. 809.
HARTLEY (Sir Harold). — **Scientific research** on the L. M. S. R. (2 400 words.)

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PIRATH. — **Eisenbahnen und Kraftwagen** im öffe lichen Verkehrswesen (10 1/4 Seiten).

1932 **656 .23 (.4**
Die Reichsbahn, Nr. 18, S. 431.
VOGT. — Welche **Tarifpolitik** muss heute die De sche Reichsbahn-Gesellschaft verfolgen ? (9 Seiten.

1932 **385 .1 (.4**
Die Reichsbahn, Nr. 19, S. 458.
ROEBE. — Die **Investierungspolitik** der Reichsba (8 Seiten.)

1932 **656 (.4**
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LEIBBRAND. — **Gegenwartsaufgaben** im Reic bahnbetrieb. Nach einem Vortrag, gehalten am März 1932 in der Eisenbahnwissenschaftlichen Wo der Verwaltungs-Akademie Berlin. (11 Seiten. Zeic Diagr. & 1 Tafel.)

1932 **621 .139 (.43). 625 .27 (.4 & 656 .237 (.4**
Die Reichsbahn, Nr. 21, S. 502.
SPIRO. — Das **Reichsbahn-Zentralamt** für Einka und seine Einkaufswirtschaft. Nach einem am 2. 1931 vor dem Berliner Bezirksverein Deutscher In nieure gehaltenen Vortrag. (10 Seiten & Diagr.)

1932 656 .1 (.43)
Reichsbahn, Nr. 22, S. 522.
Bemerkenswerte Änderungen der **Regelung des Kraftfahrzeugverkehrs.** (5 Seiten.)

Elektrische Bahnen. (Berlin.)

1932 621 .335 (.44)
Elektrische Bahnen, Juni, S. 125.
LEBOUCHER. — Die neuen **Gleichstromlokomotiven** Bauart $B_0 + B_0$. (1 500 Volt) der Midi-Eisenbahngesellschaft. (5 400 Wörter & Abb.) (Fortsetzung folgt.)

1932 621 .336
Elektrische Bahnen, Juni, S. 134.
LUSBECK (W.). — Hilfsmittel für die **Fahrleitungsunterhaltung.** (2 500 Wörter & Abb.)

1932 621 .33
Elektrische Bahnen, Juni, S. 136.
VOGEL (Th.). — Arbeitsorganisation und Zeitaufnahme des Tragseil- und Fahrdrahtzuges bei der **Elektrisierung** zweigleisiger Bahnen, auf freier Strecke und Bahnhöfen. (5 000 Wörter & Abb.) (Schluss folgt.)

1932 621 .337
Elektrische Bahnen, Juni, S. 146.
HELSING (O.). — Eine neue Sicherheitseinrichtung für die **Einmannbedienung** elektrischer Lokomotiven. (400 Wörter & Abb.)

Glasers Annalen. (Berlin.)

1932 621 .43 (.43)
Glasers Annalen, Heft 12, 15. Juni, S. 105.
NIEDERSTRASSER (L.). — Bauliche Entwicklung der **Kleinlokomotiven** der Deutschen Reichsbahn. (4 400 Wörter & Abb.)

1932 625 .232
Glasers Annalen, Heft 1, 1. Juli, S. 1.
WAGNER (G.). — **Leichtmetalltüren** für Schnellbahnwagen. (5 300 Wörter & Abb.)

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1932 625 .232 (.44)
Organ für die Fortschritte des Eisenbahnwesens, Heft 13, 1. Juli, S. 243.
VALLANCIEN (J.). — Konstruktionsgrundlagen der **4-Zug-Stahlwagen** der französischen Bahnverwaltungen. (6 200 Wörter & Abb.)

1932 625 .26 (.44)
Organ für die Fortschritte des Eisenbahnwesens, Heft 13, 1. Juli, S. 253.
HENTSCHEL. — Die **Unterhaltung der Personenwagen** in den Reichsbahnausbesserungswerken. (3 900 Wörter & Abb.)

1932 621 .138.5 (.43)
Organ für die Fortschritte des Eisenbahnwesens, Heft 14, 15. Juli, S. 263.
SCHÖNHERR (H.). — Die **Ausbesserung** von **Verbrennungstriebwagen** im Reichsbahn-Ausbesserungswerk Wittenberge. (3 800 Wörter & Abb.)

1932 621 .43 & 656 .225
Organ für die Fortschritte des Eisenbahnwesens, Heft 14, 15. Juli, S. 269.
WITTE (F.). — Die **Kleinlokomotive,** ein Betriebsmittel zur Rationalisierung des Nahgüterverkehrs. (3 400 Wörter & Abb.)

Verkehrstechnische Woche. (Berlin.)

1932 621 .133 .1
Verkehrstechnische Woche, Nr. 14, S. 225; Nr. 15, S. 240.
EBERT. — Die Wirtschaftlichkeitsgrenze für **Ruhe- und Frischfeuer der Lokomotiven.** (7 Seiten & Diagr.)

1932 625 .245 & 656 .225
Verkehrstechnische Woche, Nr. 15, S. 237.
WENTZEL. — **Grossbehälter-Verkehr.** (4 Seiten, Zeichn. & Abb.)

1932 621 .132.7
Verkehrstechnische Woche, Nr. 16, S. 249.
GALLE. — Aufgaben und Wirtschaftlichkeit der **Kleinlokomotiven im Rangierdienst.** Nach einem Vortrag, gehalten im Verein für Eisenbahnkunde am 9. Februar 1932. (7 Seiten, Zeichn., Diagr. & Abb.)

1932 625 .212
Verkehrstechnische Woche, Nr. 16, S. 256.
POCHNER. — Entwicklung und Aussichten der gesteuerten **Lenkachsen.** (3 Seiten & Zeichn.)

1932 656 (.43)
Verkehrstechnische Woche, Nr. 17, S. 261.
SARTER. — **Verkehrsfragen** von wirtschaftlicher Bedeutung für die Reichsbahn. Nach einem Vortrage auf der 3. Studienkonferenz der Deutschen Reichsbahn-Gesellschaft mit Professoren an deutschen Universitäten und Hochschulen am 8. März 1932. (10 Seiten.)

1932 625 .23 (.43)
Verkehrstechnische Woche, Nr. 19, S. 294.
STROEBE. — Technische Fortschritte im **Personenwagenbau** bei der Reichsbahn. (3 Seiten & Abb.)

1932 656 .212.6
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RANDZIO. — **Förderkostengleichungen.**
I. Die Schubkarrenförderung. Zeitverlust = n^t.
II. Förderung mit zweirädrigen Handkippkarren.
III. Zweirädrige Pferdekippkarren.
IV. Gleisförderung :
a) mit Handbetrieb ;
b) Gleisförderung mit Pferden ;
c) Gleisförderung mit leichter Lokomotive, z. B. 10-20 PS-Benzol-Lok. ;
d) Gleisförderung mit schwerer Dampf-Lokomotive, 200 PS.
(5 Seiten.)

1932 **385 .15 (.496 + .56)**
Verkehrstechnische Woche, Nr. 21, S. 317; Nr. 22, S. 333.

EMMELIUS. — Der **Übergang zum Staatsbahnsystem** in der Türkischen Republik und die Entwicklung der Türkischen Reichsbahn. (8 Seiten & 1 Karte.)

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1932 **62. (01**
Zeitschrift des Ver. deutsch. Ing., Nr. 27, 2, Juli, S. 659.

SIEBEL (E.) & ULRICH (M.). — Die Bestimmung von Zeit-Dehngrenzen im **Dauerstandversuch.** (4 000 Wörter & Abb.)

1932 **624 .62 (.73)**
Zeitschrift des Ver. deutsch. Ing., Nr. 27, 2, Juli, S. 665

BOOS (W.). — Die **Kill van Kull-Brücke** in New York. (1 200 Wörter & Abb.)

1932 **621 .392**
Zeitschrift des Ver. deutsch. Ing., Nr. 28. 9 Juli,, S. 679.

CZTERNASTY (G.). — **Festigkeitseigenschaften** einer hochwertigen Lichtbogenschweissung. (2 100 Wörter & Abb.)

1932 **62. (01**
Zeitschrift des Ver. deutsch. Ing., Nr. 28. 9 Juli,, S. 683.

LUDWIK (P.) & SCHEU (R.). — Die Veränderlichkeit der **Werkstoffdämpfung.** (2 900 Wörter & Abb.)

1932 **621 .43 (.43)**
Zeitschr. des Ver. deutsch. Ing., Nr. 29, 16. Juli, S. 705.

MAYER (M.). — Fortschritte und Aussichten des **Diesellokomotivbaus** unter besonderer Berücksichtigung der Diesel-Druckluftlokomotive der Deutschen Reichsbahn. (3 300 Wörter & Abb.)

1932 **669 .1**
Zeitsch. des Ver. deutsch. Ing., Nr. 30, 23. Juli, S. 729.

EILENDER (W.) & KIESSLER (H.). — Beitrag zur Frage der **Flockenbildung in Baustählen.** (3 800 Wörter & Abb.)

Zeitung des Vereins deutscher Eisenbahnverwaltungen. (Berlin.)

1932 **625 .144**
Zeitung des Vereins deutscher Eisenbahnverwaltungen, Nr. 26, 30. Juni, S. 561.

SEEGER. — **Vergrösserung des Gleisabstandes** auf normalspurigen zweigleisigen Strecken. (2 400 Wörter & Abb.)

1932 **3**
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BLUM. — Grundprobleme des **städtischen Verkeh** Abgekürzte Wiedergabe eines im Kurs über neuzeitlic Verkehrsprobleme des Schweizerischen Ingenieur- u Architektenvereins in Zürich gehaltenen Vortrag (10 000 Wörter.)

1932 **656 .211 & 656 .21**
Zeitung des Vereins deutscher Eisenbahnverwaltunge Nr. 28, 14. Juli, S. 597.

KÜMMELL. — Anlage von **Bahnhöfen,** Abfertigu und Umladehallen. (4 300 Wörter & Abb.)

1932 **385 .582 (.43**
Zeitung des Vereins deutscher Eisenbahnverwaltunge Nr. 29, 21. Juli, S. 620.

MARTENS (A.). — Erfolgreiche **Unfallverhütung** i den Reichsbahn-Ausbesserungswerken. (2 900 Wörte & Abb.)

In English.

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1932 **621 .132.3 (.73**
Engineer, No. 3989, June 24, p. 684.

BREWER (F. W.). — **Three-cylinder express en gines** : Philadelphia and Reading Railroad. (2 300 word & fig.)

1932 **621 .134.1 (.42)**
Engineer, No. 3989, June 24, p. 691.

London & North Eastern Railway. **Tank engine wit booster.** (1 000 words & fig.)

1932 **669 .1**
Engineer, No. 3989, June 24, p. 702.

Stainless iron and steel production. (3 400 words.)

1932 **625 .143.3**
Engineer, No. 3990, July 1, p. 2.

HEARN (Sir Gordon). — **Wear of rails on a curve.** (4 300 words & fig.)

1932 **621 .7 & 669 .1**
Engineer, No. 3990, July 1, p. 5.

Machinability of steel. (4 400 words & fig.)

1932 **656 .1 & 656 .2**
Engineer, No. 3990, July 1, p. 13.

Rail and road transport. (1 500 words.)

1932 **621 .335 (.494)**
Engineer, No. 3990, July 1, p. 21.

Swiss Railways. — **8 800 H. P. electric locomotive.** (900 words & fig.)

1932 **621 .33 (.42)**
Engineer, No. 3991, July 8, p. 28.

Brighton (Southern Railway) **electrification scheme.** (2 200 words & fig.)

932 656 .254 (.42)
ineer, No. 3991, July 8, p. 44.
tension of **teleprinter** circuits on the London & h Eastern Ry. (500 words.)

932 621 .9
ineer, No. 3992, July 15, p. 70.
new **hand-controlled rotary scaling hammer.** (500 ds.)

Engineering. (London.)

932 62. (01 & 669
ineering, No. 3467, June 24, p. 731.
HRAN (W.). — A machine for **determining the fa-** of metals. (4 000 words & fig.)

932 621 .86
ineering, No. 3467, June 24, p. 740.
ectrically-operated traverser for railway rolling . (900 words & fig.)

932 621 .392
ineering, No. 3467, June 24,, p. 755.
nall portable welding set. (350 words & fig.)

932 624 .8 (.489)
ineering, No. 3468, July 1, p. 1.
NGELUND (A.). — **Highway and railway bridge ss Alssund,** Denmark. (3 800 words & fig.)

932 621 .335 (.494)
neering, No. 3468, July, p. 22.
800-HP. single-phase electric locomotive for the ss Federal Railways. (1 000 words & fig.)

932 621 .392 (.42) & 721 .9 (.42)
ineering, No. 3468, July 1, p. 23.
ritish practice in the welding of structural steel. 0 words.)

932 621 .132.3 (.73) & 621 132.5 (.73)
ineering, No. 3469, July 8, p. 35.
5-2 type and 4-8-2 type locomotives on the Penn- ania Railroad. (1 800 words & fig.)

932 621 .33 (.42)
ineering, No. 3469, July 8, p. 38.
electrification on the Southern Railway. (2 200 ds & fig.)

932 62. (01 (.42)
ineering, No. 3469, July 8, p. 53.
lf-indicating universal testing machine. (1 800 ds & fig.)

932 621 .3 (06
ineering, No. 3470, July 15, p. 73.
he International Electrical Congress in Paris. (4 500 ds.)

932 389
ineering, No. 3470, July 15, p. 79.
he Microlux Optical micrometer. (750 words & fig.)

1932 621 .31 (08 (.73)
Engineering, No. 3470, July 15, p. 79.
The **production and transformation of electrical energy in the United States.** (3 200 words & 2 tables.)

Engineering News-Record. (New York.)

1932 624 & 693
Engineering News-Record, No. 24, June 16, p. 851.
ARMOUR (C. V.). — **Thick concrete slab acts well under locomotive test load.** (900 words & fig.)

1932 624
Engineering News-Record, No. 25, June 30, p. 911.
Guard-rail practice varies on railway bridges. (1 000 words.)

1932 656 .211.5 (.73)
Engineering News-Record, No. 25, June 30, p. 912.
PAYZANT (O. S.). — **Long-span platform roofs** for Cincinnati station. (500 words & fig.)

1932 625 .13 (.73)
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Boston-East Boston **vehicle tunnel.** Design and construction. I. — (3 100 words & fig.)

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Engineering News-Record, No. 25, June 30, p. 923.
Material specifications and research at American Society for Testing Materials' meeting. (6 300 words.)

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1932 385 .1 (.44)
Indian Railway Gazette, June, p. 137.
The **French railway crisis.** (2 400 words.)

1932 621 .335 (.43) & 621 .43 (.43)
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STRAUSS (F.). **Heavy oil and Diesel electric traction in Germany. Restricting competition of road transport.** (1 300 words.)

London & North Eastern Railway Magazine. (London.)

1932 625 .15 (.42)
London & North Eastern Ry. Mag., July, p. 358.
THOM (R. A.). — **Manufacture of points and crossings** at Gorton Works. (1 000 words & fig.)

Modern Transport. (London.)

1932 385 .1 (.94)
Modern Transport, No. 693, June 25, p. 2.
Australian transport. (1 100 words.)

1932 **621 .13 (0 (.42)**
Modern Transport, No. 693, June 25, p. 3.
Progress in locomotive design. (4 400 words & fig.)

1932 **313 .385 (.71)**
Modern Transport, No. 693, June 25, p. 5.
LIDDY (S. J. W.). — **Statistical methods** on the Canadian Pacific Railway. (2 300 words & fig.)

1932 **621 .43**
Modern Transport, No. 693, June 25, p. 7.
Oil engines for rail traction. (1 100 words & fig.)

1932 **621 .338 (.493)**
Modern Transport, No. 694, July 2, p. 3.
Light railways in Belgium. (2 100 words & fig.)

1932 **656 .253 (.42)**
Modern Transport, No. 694, July 2, p. 11.
Fog-penetrating lamps for railway signalling. (500 words.)

1932 **621 .138.2 (.42)**
Modern Transport, No. 695, July 9, p. 3.
Mechanical coaling of locomotives (2 400 words & fig.)

1932 **656 .235 .6**
Modern Transport, No. 695, July 9, p. 5.
Industrial traffic management. — No. 25. **Railway companies and demurrage.** (1 900 words.)

1932 **621 .33 (.42)**
Modern Transport, No. 695, July 9, p. 7.
Electrification of the Southern Railway. — Completion of main line section to Three Bridges. (4 000 words, 1 table & fig.)

1932 **621 .335 (.41) & 621 .43 (.41)**
Modern Transport, No. 696, July 16, p. 5.
Rail cars for Ireland. — Great Northern Railway development. (750 words & fig.)

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1931 **624 .2**
Proc., Inst. of Civil Eng., selected engineering paper, No. 109.
MONTAGNON (E. L.). — **The effect of adding flange-plates to plate-web girders.** (4 400 words & fig.)

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BENNETT (R.). — **Pile-driving and the supporting capacity of piles.** (3 100 words & fig.)

1931 **621 .31 & 627**
Proc., Inst. of Civil Eng., selected engineering paper, No. 115.
SIEVEKING (H. A.). — **Hydraulic accumulator stations,** with special reference to conditions affecting their use in Great Britain. (5 000 words & fig.)

1931 **625 .14**
Proc., Inst. of Civil Eng., selected engineering pap No. 116.
INGLIS (R. A.). — **The proportioning of railw track fishplates** from an economic standpoint. (3 words & fig.)

1931 **621 .31 (.7**
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LEGGET (R. F.). — **Winter construction of an a tomatic low-head hydro-electric plant** in Canada. (8 words, 1 table & fig.)

1931 **624**
Proc., Inst. of Civil Eng., selected engineering pap No. 119.
BLACKADDER (W.). — A direct method for t construction of **influence lines for continuous girde** (5 300 words.)

Proceedings, Institution of Railway Signal Engineers. (Reading).

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BUCKINGHAM (W.) & WEIR (A.). — **Indicato and repeaters.** (9 300 words & fig.)

1932 **656 .2**
Proc., Inst. Ry. Signal Eng., Part II, p. 267.
WAGENRIEDER (B. F.). — **Disc and miniature s nals for shunting and setback movements.** (9 8 words & fig.)

1932 **656 .2**
Proc., Inst. Ry. Signal Eng., Part II, p. 292.
MYERS (H.). — **Automatic train control.** (5 8 words.)

1932 **656 .25 (.5**
Proc., Inst. Ry. Signal Eng., Part II, p. 308.
DAVIES (C. E.). — **Developments in signalling** the Bombay suburban section of the Bombay, Baro and Central India Railway since 1925. (9 000 words fig.)

Railway Age. (New York.)

1932 **621 .13 (0 (.7**
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BINKERD (R. S.). — **What modern locomotives c do for American Railroads.** (1 900 words & fig.)

1932 **621 .139 (.73), 625 .18 (.7**
625 .26 (.7
Railway Age, No. 25, June 18, p. 1011.
The Railway Supply Officers Report. (4 600 words fig.)

1932 **625 .143.3 (.7**
Railway Age, No. 25, June 18, p. 1017.
MORGAN (H. H.) & MOONEY (J. R.). — **Head fa ures in intermediate manganese rail.** (1 900 words fig.)

1932 **625 .143. (0**
ilway Age, No. 26, June 25, p. 1059.
ALBOT (A. N.). — **Why high stresses occur in** ls. (2 700 words.)

1932 **621 .39 (06 (.73), 625 .18 (06 (.73) & 625 .26 (06 (.73)**
ilway Age, No. 26, June 25, p. 1062.
Railway Supply Officers hold annual convention. 200 words.)

1932 **621 .13 (06 (.73) & 621 .2 (06 (.73)**
ilway Age, No. 26, June 26, p. 1072.
Mechanical Division, American Railway Association, lds its **annual meeting** in Chicago. (17 000 words.)

1932 **656 .1 (.73)**
ilway Age, No. 26, June 26, p. 1084.
How should railways provide for **bus substitutions ?** 000 words & fig.)

1932 **656 .1 (.73) & 656 .261 (.73)**
ilway Age, No. 1, July 2, p. 7.
R. R. **motor operations** reviewed. (3 500 words.)

1932 **614 .7 (.71 + .73) & 621 .133.4 (.71 + .73)**
ilway Age, No. 1, July 2, p. 13.
Smoke abatement and fuel economy. (2 800 words.)

Railway Engineer. (London.)

1932 **625 .23**
ilway Engineer, July, p. 245.
Railway carriage and wagon design and equipment. series of 39 short articles). (36 000 words & fig.)

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ilway Engineering and Maintenance. (Chicago.)

1932 **625 .142.2**
ilway Engineering & Maintenance, July, p. 440.
How long do **treated ties last ?** (3 500 words & fig.)

1932 **621 .392 (.73)**
ilway Engineering & Maintenance, July, p. 444.
Use **arc welding in strengthening and repairing metal** idges. (3 700 words & fig.)

1932 **693 (.73)**
ilway Engineering & Maintenance, July, p. 451.
Ready-mixed concrete finds place in B. & B. work. 900 words & fig.)

1932 **625 .14 (.44)**
ilway Engineering & Maintenance, July, p. 453.
HEARN (Sir Gordon). — Is France ahead in **track** actice. (2 400 words & fig.)

Railway Gazette. (London.)

1932 **625 .143.1 (.68)**
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New standard section for main line use on the 3 ft. 6 in. gauge, South African Railways. (450 words & fig.)

1932 **625 .236 (.45)**
Railway Gazette, No. 26, June 24, p. 903.
New carriage-cleaning shed, Italian State Rys. (200 words & fig.)

1932 **656 .211 (.44) & 725 .31 (.44)**
Railway Gazette, No. 26, June 24, p. 905.
HARRAND (G.). — **The reconstruction of the Gare de l'Est, Paris.** (2 400 words & fig.)

1932 **656 .225 (.42)**
Railway Gazette, No. 1, July 1, p. 9.
Fast goods train services on the L. M. S. Ry. (2 000 words & fig.)

1932 **656 .261 (.42)**
Railway Gazette, No. 1, July 1, p. 14.
A modern railway **goods station.** (1 500 words & fig.)

1932 **621 .132.6 (.54)**
Railway Gazette, No. 1, July 1, p. 19.
Standard tank locomotives for Indian Railways. (900 words & fig.)

1932 **621 .392 (.42) & 624 (.42)**
Railway Gazette, No. 1, July 1, p. 25.
All-welded steel bridges. (700 words & fig.)

1932 **621 .132.6**
Railway Gazette, No. 1, July 1, p. 48.
A modern **single wheeler tank locomotive.** (450 words & fig.)

1932 **621 .33 (.42)**
Railway Gazette, No. 1, July 1, p. 51.
First **main line electrification** in England. (1 900 words & fig.)

1932 **656 .215**
Railway Gazette, No. 1, July 1, p. 54.
Station lighting. (1 500 words & fig.)

1932 **656 .211.5 (.481 + .489)**
Railway Gazette, No. 2, July 15, p. 75.
Danish **train ferry.** (500 words & fig.)

1932 **621 .93**
Railway Gazette, No. 2, July 15, p. 76.
Improved **sawing off machines.** (500 words & fig.)

1932 **625 .232 (.42)**
Railway Gazette, No. 2, July 15, p. 76.
Café cars on the Great Western Railway. (250 words & fig.)

1932 **625 .24 (.67)**
Railway Gazette, No. 2, July 15, p. 79.
New **rolling stock** for the Benguela Railway. (500 words & fig.)

1932 **621 .43**
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MIALL (S.). — **Transmissions for Diesel locomotives and railcars.** The S. L. M. transmission. (1 000 words fig.)

1932 **621 .43 (.41)**
Railway Gazette, No. 2, July, p. 84.
Light Diesel railcars for Ireland. (700 words & fig.)

1932 **621 .43 (.42)**
Railway Gazette, No. 2, July 15, p. 86.
Allen railway traction Diesel engine. (1 000 words & fig.)

Railway Magazine. (London.)

1932 **656 .222.1 (.42)**
Railway Magazine, July, p. 10.
ALLEN (C. J.). — **A super-speed record.** (1 200 words, 3 tables & fig.

Railway Mechanical Engineer. (New York.)

1932 **621 .13 (06 (.73) & 625 .2 (06 (.73)**
Railway Mechanical Engineer, July, p. 259.
Mechanical Division, American Railway Association, holds two-day **meeting** at Chicago. (30 000 words.)

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1932 **385 .586 (.68)**
South African Rys. & Harb. Mag., June, p. 715.
Adams (E. R.). — Some notes on the **training of apprentices** on the South African Railways. (2 200 words & fig.)

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1932 **621 .392**
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WALKER (F. B.). — **Quality of welds** may be determined by visual inspection. (1 400 words & fig.)

1932 **625 .255**
Transit Journal, July, p. 311.
DAVIS (H. A.). — **Magnetic track brakes** used for emergency stops on Indiana Railroad cars. (600 words & fig.)

1932 **625 .248 (.42)**
Transit Journal, July, p. 317.
Progressive car cleaning and inspection at London. (1 800 words & fig.)

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University of Illinois Bulletin, No. 45, February 2, p
KETCHUM (M. S.) & DRAFFIN (J. O.).— **Streng** of light I beams. (7 200 words, 17 tables & fi

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In Spanish.

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1932 **621 .335 (.46**
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PONTECORVO (L.). — **Electrificación** de los Ferr carriles Vascongados. (4 000 palabras & fig.) (Con nuará.)

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AZA (P.) & COSTILLA (B.). — ¿ El « **tren medio** como índice de la explotación en los ferrocarriles (2 400 palabras, 2 cuadros & fig.)

1932 **625 .245 & 656 .22**
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WOHL (P.). — El progreso del « **container** » e América y Europa. (5 400 palabras & fig.)

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(1) The numbers placed over the title of each book are those of the decimal classification proposed by the Railway Congress conjointly with the Office Bibliographique International, of Brussels. (See « Bibliographical Decimal Classification as applied to Railway Science », by L. WEISSENBRUCH, in the number for November, 1897, of the *Bulletin of the International Railway Congress*, p. 1509).

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MARCOTTE (E.). — Un bon **béton.** (6 000 mots & .)

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1932 625 .245 & 656 .225

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WAETJEN (C. H.). — Emploi des « **containers** » our transports combinés par rail et par route. (800 ots & fig.)

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1932 385 .1 (.43)

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1932 621 .335 (.44)

énie Civil, nº 2611, 27 août, p. 197 ; nº 2612, 3 septembre, p. 226 ; nº 2613, 10 septembre, p. 248.

COQUET (H.). — La **traction électrique** sur les Che ins de fer du Midi. Locomotives B-B, séries E 4101 et E 4601. (19 200 mots & fig.) (A suivre.)

1932 62. (01

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ROY (L.). — De la ligne élastique aux formules fondamentales de la **résistance des matériaux.** (5 000 mots & fig.)

1932 62. (01 & 721 .1

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CAUFOURIER (P.). — La poussée des terres et le calcul des **murs de soutènement.** Calculs et expériences de M. C. F. Jenkin. (4 700 mots & fig.)

1932 69

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Le contrôle de la sécurité dans les **constructions.** (1 900 mots.)

1932 621 .89

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1932 625 .235

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D'AUVIGNY (H.). — L'emploi des **alliages légers** dans la construction du matériel roulant. (3 700 mots & fig.)

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1932 656 .1

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1932 625 .13 (.45)

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MARIVAL (J.). — Le plus long **tunnel** du monde à deux voies vient d'être achevé en Italie. (2 900 mots & fig.)

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1932 621 .43 (.44)

L'équipement rural, juillet, p. 55.

MENETRIER (A.). — Les **autorails** de la Compagnie des chemins de fer du Midi français. (2 700 mots & fig.)

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1932 621 .132.8 & 621 .43

Les Chemins de fer et les Tramways, septembre, p. 147.

SPIESS (E.). — **Locomotives à combustion interne.** (10 000 mots & fig.)

1932 **625 .212**

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MERCOYROL (A.). — **Graisseur pour boudins de roues** de véhicules circulant sur rails à ornière ou autres plus spécialement pour les véhicules des tramways. (2 900 mots & fig.)

1932 **656 .254**

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Dispositif d'actionnement d'un **porte-pétard** électrique de voie unique. (1 100 mots & fig.)

L'Industrie des voies ferrées et des transports automobiles. (Paris.)

1932 **625 .62 (06**

L'Ind. des voies ferrées et des transp. autom., août, p. 221.

BORDAS (F.). — 23e **Congrès International** de Tramways, de Chemins de fer d'intérêt local et de Transports publics automobiles. (La Haye, 26 juin-4 juillet 1932.) (8 000 mots & fig.)

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1932 **621 .132.3 (.44)**

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PARMANTIER (A.). — La **locomotive 241.C-1** à grande vitesse (type 2-4-1) de la Compagnie des Chemins de fer de Paris à Lyon et à la Méditerranée. (7 500 mots & fig.)

1932 **385 .113 (.44)**

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Les **résultats de l'exploitation** des cinq grandes compagnies de chemins de fer en 1931. (10 000 mots & diagr.)

1932 **385 .113 (.42)**

Revue générale des chemins de fer, septembre, p. 261.

Les **résultats d'exploitation** des chemins de fer britanniques en 1930 et 1931. (6 100 mots.)

1932 **621 .134.2 (.44)**

Revue générale des chemins de fer, septembre, p. 273.

Tiroir cylindrique de distribution à double admission et double échappement, système « Willoteaux » de la Compagnie du Chemin de fer de Paris à Orléans. (1 300 mots & fig.)

1932 **385 (.6)**

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Le **Transcontinental** Sud-Nord africain. (700 mots & carte.)

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BOURGOIN. — **L'électrification** des réseaux français et leur déficit financier. (6 000 mots.)

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GENEST. — Die **öffentlichen Abgaben** (Steuern, ... bühren und Beiträge) der Deutschen Reichsbahn-Ges... schaft. (14 800 Wörter.)

1932 **656 .1 (.73) & 656 .2 (.**

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MERKERT (E.). Die Schlussfolgerungen des B... desverkehrsamts der Vereinigten Staaten von Amer... über den **Eisenbahn- und Kraftwagenverkehr.** (3 ... Wörter.)

1932 **3**

Archiv für Eisenbahnwesen, Juli-August, S. 891.

LEHNER (F.). — Die **Linienführung** innerstädtisch... Verkehrsmittel. (5 300 Wörter, 1 Tafel & Abb.) (Fo... setzung folgt.)

1932 **656 .2**

Archiv für Eisenbahnwesen, Juli-August, S. 917.

GIESE (K.). — Der **Hafenbahnvertrag** in Theorie u... Praxis. Zur Frage der Organisation des Hafenbahnb... triebs. (18 600 Wörter & Abb.) (Schluss.)

1932 **385 .113 (.49**

Archiv für Eisenbahnwesen, Juli-August, S. 999.

VON RENESSE. — Die **nationale Gesellschaft d... belgischen Eisenbahnen** im vierten Geschäftsjahr (1. J... nuar bis 31. Dezember 1930), dargestellt auf Gru... des Geschäftsberichts der Gesellschaft und des Berich... des Verwaltungsrats. (5 300 Wörter.) (Schluss.)

1932 **385 .113 (.437**

Archiv für Eisenbahnwesen, Juli-August, S. 1021.

Die **Eisenbahnen** der tschechoslowakischen Republi... in den Jahren 1929 und 1930. (6 100 Wörter.)

1932 **385 .113 (.494**

Archiv für Eisenbahnwesen, Juli-August, S. 1043.

NITSCHKE. — Die **Schweizerischen Bundesbahne...** im Jahr 1930. (1 500 Wörter.)

1932 **313 .385 (.41**

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Die **Eisenbahnen** Irlands 1929 und 1930. (3 000 Wör... ter.)

Die Lokomotive. (Wien).

1932 **621 .132.1 (.436)**

Die Lokomotive, September, S. 159.

STEFFAN. — Die letzten **Lokomotiven** aus der Maschinenfabrik der Staatseisenbahngesellschaft in Wien, IV. Teil. (3 800 Wörter & Abb.)

Elektrische Bahnen. (Berlin.)

1932 **625 .234 (.433)**

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AUCH (A.). — **Elektrische Zugvorheizanlage** im ptbahnhof München. (3 300 Wörter & Abb.)

1932 **621 .33 (.73)**

ktrische Bahnen, Heft 8, August, S. 181.

RIGHT (G. I.). — Der **elektrische Zugbetrieb** der ding Company auf den Vorortbahnen von Philadel-. (1 500 Wörter & Abb.)

1932 **625 .4 (.431) & 621 .332 (.431)**

ktrische Bahnen, Heft 8, August, S. 192.

EINRICH (R.). — Die **Stromversorgungsanlagen** die Nebenbetriebe in den Unterwerken der Berliner ahn. (2 400 Wörter & Abb.)

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1932 **621 .91 & 621 .138.5**

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ÜTER. — Das **Schleifen der Lokomotiv-Rahmen-schnitte** auf der stationären Schmaltz'schen Rahmen-kenschleifmaschine. (2 900 Wörter & Abb.)

1932 **625 .23 (0**

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Mess- und Wiegevorrichtung für **Drehgestell-Wagen-ten.** (1 700 Wörter & Abb.)

1932 **625 .216**

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ORENZ (R.). — Ein neues Übergangsverfahren und ne Bedeutung für die europäische **Kupplungsfrage.** 500 Wörter & Abb.)

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1932 **388 (.43) & 656 .22 (.43)**

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MÜLLER (W.). — Netztafeln für die Untersuchung s **Betriebes** der Berliner Stadtbahn. (6 300 Wörter & bb.)

1932 **625 .14 & 625 .142.4**

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NEMESEK (J.). — Das **Langschienengleis** auf Eisen-tonschwellen. (2 900 Wörter & Abb.)

1932 **656 .254**

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CANTZ (H.). — **Schrankenbeleuchtung** durch Schein-erfer. (600 Wörter & Abb.)

1932 **625 .215**

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HEUMANN. — **Bogenlauf** vierachsiger Eisenbahn-agen. (5 700 Wörter & Abb.) (Schluss folgt.)

1932 **621 .133.3**

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RITTER (P.). — Einfluss der freien Rohrdurchgangs-fläche auf die **Leistung des Lokomotivkessels.** (1 800 Wörter & Abb.)

1932 **625 .216**

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HALFMANN. — Neuere **Pufferfedern** für Fahrzeuge der Deutschen Reichsbahn-Gesellschaft. (700 Wörter & Abb.)

1932 **621 .133.1**

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Kohlenstaublokomotive. (400 Wörter & Abb.)

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JOOSTEN. — Der Bau der **Donaubrücke** bei Belgrad. (2 500 Wörter & Abb.)

1932 **62. (01 & 721 .9**

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HARRY GOTTFELDT. — Querschnitt und Umriss. — **Stahlbau.** (6 500 Wörter & Abb.)

1932 **621 .87 (.431)**

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WUNDRAM (G.). — Neuartige **Doppelwippkrane** im Hamburger Hafen. (2 200 Wörter & Abb.)

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THUM (A.) & OCHS (H.). — Die **Bekämpfung der Korrosionsermüdung** durch Druckvorspannung. (1 400 Wörter & Abb.)

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1932 **656 .259**

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SCHMIDT (A.). — Radaufhalter, ein neues **Rangier-hilfsmittel.** (2 400 Wörter & Abb.)

1932 **625 .143.4**

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SALLER. — Zur Beseitigung der **Schienenstossfrage.** (900 Wörter & Abb.)

1932 **656 .1 & 656 .2**
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1932 **385. (07 (.43) & 625 .245 (.43)**
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SCHRÖDER. — Die **Unterrichtswagen** der Deutschen Reichsbahn. (3 300 Wörter & Abb.)

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1932 **621 .33**
Engineer, No. 3996, August 12, p. 152; No. 3997, August 19, p. 173.
WATSON (H.). — The economics of **urban electric railways.** New **surface railways.** (3 500 words & tables.)

1932 **621 .13 (.093)**
Engineer, No. 3996, August 12, p. 154.
DENDY MARSHALL (C. F.). — Links in the **history of the locomotive.** (2 200 words & fig.)

1932 **621 .138.2 (.42)**
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New **coaling plant** for the London & North Eastern Railway, Hull. (900 words.)

1932 **621 .31 (.42)**
Engineer, No. 3997, August 19, p. 174; No. 3998, August 26, p. 201.
The Iron bridge **power station.** (7 300 words & fig.)

1932 **656 .1 (.42) & 656 .2 (.42)**
Engineer, No. 3997, August 19, p. 178.
Rail and road transport. (7 000 words.)

1932 **621. 14**
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The one-man « Sentinel » **steam wagon.** (400 words & fig.)

1932 **669 .1**
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GRAHAM (H. W.). — A modern conception of **steel.** (5 000 words.)

1932 **621 .116 & 621 .392**
Engineer, No. 3998, August 26, p. 214.
Electrically-welded boiler steam drums. (500 words & fig.)

1932 **656 .212.6 (.42)**
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Ship **coaling plant** at Dover. (3 300 words & fig.)

1932 **625**
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Springs for rolling stock. (2 400 words.)

1932 **625 .4 & 656**
Engineer, No. 3999, September 2, p. 239.
Safety on tube railways. (700 words.)

1932 **624 .32 (.59**
Engineer, No. 3999, September 2, p. 248.
Bangkok **bridge.** (3 300 words & fig.)

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1932 **62. (**
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The pomp-enders short-time creep test. (3 000 wor & 6 tables.)

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Vertical-lift bridges over the Hackensack River, N Y., U. S. A. (1 500 words & fig.)

1932 **621 .8**
Engineering, No. 3476, August 26, p. 242.
5-ton **steam-travelling crane.** (2 200 words & fig.)

1932 **347 .763 (.42) & 656 .1 (.42**
Engineering, No. 3476, August 26, p. 247.
The **control of road transport.** (3 300 words.)

1932 **621 .**
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Direct-comparison contour projections. (1 000 words & fig.)

1932 **62**
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Bridgework. (2 800 words.)

1932 **62. (01 & 669 .**
Engineering, No. 3477, September 2, p. 280.
LEA (F. C.). — The **strength of materials** as affected by discontinuities and surface conditions. (1 600 words & fig.)

1932 **62. (01**
Engineering, No. 3477, September 2, p. 285.
The Firth **hardometer,** (600 words.)

1932 **621 .43**
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DAVIES (S. J.). — The Michel **high-speed oil engine** and its performance. (4 800 words & fig.)

1932 **621 .33 (.42)**
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The Barking-Upminster **electrification.** (2 200 words.)

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TERZAGHI (K.). — **Soil-loading test practice in Europe.** (1 600 words.)

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CASAGRANDE (A.). — **Compressibility of clay increased.** (1 700 words.)

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STEPHENSON (H. K.) & FEINGOLD (E. B.). — **Foundation soil overloading shown by penetration test.** (200 words & fig.)

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1932 621 .132.5 (.44)
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Four-cylinder compound freight locomotive, P. L. M. Rys. (1 700 words & fig.)

1932 621 .132.1 (.437)
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Modern locomotives of the Czechoslovakian State Rys. (1 700 words & fig.)

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PHILLIPSON (E. A.). — **Steam locomotive design : data and formulæ.** (2 000 words & fig.)

1932 656 .211.4 (.44)
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The Paris terminus of the Eastern Railway of France. (1 900 words & fig.)

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Electric locomotive design : V. (900 words, 2 tables fig.)

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PERRY (Th. D.). — **Plywood : a product of unusual possibilities.** (2 700 words & fig.)

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CHRISTIE (A. G.). — **Central-station plant and performance.** (3 000 words & fig.)

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FIRESTONE (F. A.). — **Inspection of surfaces for minute defects.** (700 words & fig.)

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A logical report **(rail and road).** (1 100 words.)

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GÜNTHER (K.) & SOLVEEN. — **Testing equipment for locomotives.** (1 600 words & fig.)

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Modern Transport, No. 701, August 20, p. 5.
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1932 621 .43
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New heavy-oil engine. (1 100 words & fig.)

1932 625 .144.4 (.42)
Modern Transport, No. 701, August 20, p. 7.
Mechanical relaying of permanent way. (1 100 words & fig.)

1932 656 .1 (.42)
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Conflicting views on Salter report **(rail and road).** (1 100 words).

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GRIFFITH-BOSCAWEN (Sir A.). — The conference and the Royal Commission **(rail and road).** (1 600 words.)

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Speed signalling on the London Midland & Scottish Ry. (4 000 words & fig.)

1932 621 .132.8 (.498)
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British-built **railcars** for Rumania. (1 600 words & fig.)

1932 656 .222.5
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Railways need frequent services. The case for the **rail-bus.** (2 300 words.)

1932 621 .132.5 (.47)
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Freight locomotives for Russia. — Seven coupled axles. (600 words, 1 table & fig.)

1932 **621 .33 (.42)**
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Electrification on the London Midland & Scottish Railway. (2 400 words & fig.)

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TRITTON (Sir Seymour B.). — **Railway traction by steam power.** (1 900 words.)

1932 **656 .23 (0 (.42)**
Modern Transport, No. 704, September 10, p. 7.
Problem of **passenger fares.** (12 000 words & tables.)

1932 **656 .253 (.42)**
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Signalling development on the Metropolitan Railway. New installation at Wembley Park. (2 500 words & fig.)

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1932 **624 .51 (.73)**
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AMMANN (O. H.). — George Washington **bridge** : general conception and development of design. (21 000 words & fig.)

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SINGSTAD (O.), O' ROURKE (J. F.) & HATCH (H. H.). — Fulton Street, East River **tunnels,** New York, N. Y. (3 900 words & fig.)

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1932 **625 .234 (.73)**
Railway Age, No. 5, July 30, p. 140.
Illinois Central **air conditions** its « Daylight Special ». 2 300 words & fig.)

1932 **625 .27 (.73)**
Railway Age, No. 5, July 30, p. 143.
STUDER (L. L.). — Missouri Pacific changes **storehouse methods.** (1 400 words & fig.)

1932 **721 .1 & 721 .3**
Railway Age, No. 5, July 30, p. 147.
Difficult building column changes made successfully. (1 900 words, 1 table & fig.)

1932 **656 .255 (.73)**
Railway Age, No. 5, July 30, p. 149.
Stewart's **train order delivery staff.** (300 words & fig.)

1932 **385 .4 (.73)**
Railway Age, No. 5, July 30, p. 150.
Four-system plan approved. (12 500 words.)

1932 **656 .261 (.73**
Railway Age, No. 6, August 6, p. 175.
Railroads use hundreds of **autos to cut costs.** (3 50 words & fig.)

1931 **621 .133.7 (.73**
Railway Age, No. 6, August 6, p. 181.
What **treatment for water ?** (2 100 words & fig.

1932 **621 .335 & 621 .4**
Railway Age, No. 6, August 6, p. 183.
WALKER (E. B.). — The possibilities of **gas-electri locomotives.** (1 800 words & fig.)

1932 **656 .256 .**
Railway Age, No. 6, August 6, p. 185.
ZANE (W. F.). — Applying **automatic interlocking** (1 900 words & 1 table.)

1932 **656 .1 (73**
Railway Age, No. 7, August 13, p. 210.
DUNCAN (C. S.). — **Who pays for the highways** (6 200 words & fig.)

1932 **625 .1 (.73)**
Railway Age, No. 7, August 13, p. 217.
Erie completes large **track elevation project** through Paterson, N. Y. (2 300 words & fig.)

1932 **625 .232 (.73)**
Railway Age, No. 7, August 13, p. 221.
Long Island has **double-tier coach in trial in suburban service.** (1 200 words & fig.)

1932 **656 .257 (06 (.73)**
Railway Age, No. 7, August 13. p. 227.
Railway Accounting Officers Association meet at Buffalo, N. Y., on August 2-4, 1932. (6 000 words.)

1932 **725 .31 (.71)**
Railway Age, No. 8, August 20, p. 247.
Attractive **passenger stations.** (2 800 words & fig.)

1932 **625 .244 (.73)**
Railway Age, No. 8, August 20, p. 250.
The « **Frigi Warm** » **car dividing curtain.** (1 700 words & fig.)

1932 **621 .139 (06 (.73), 625 .18 (06 (.73) & 625 .27 (.73)**
Railway Age, No. 8, August 20, p. 252.
FARRELL (W. J.). — The **Purchases and Stores Association reviews its work.** (2 800 words & fig.)

1932 **625 .214 (.73)**
Railway Age, No. 8, August 20, p. 255.
Roller-bearings performance on locomotives. (4 500 words & fig.)

1932 **656 .237.3 (.73)**
Railway Age, No. 8, August 20, p. 259.
Revised **waybills** increase revenue. (1 500 words.)

1932 656 .255 (.73)
ailway Age, No. 8, August 20, p. 263.
Milwaukee reverts to single track on 200 miles of ne. (2 600 words & fig.)

1932 625 .123
ailway Age, No. 9, August 27, p. 282.
Taking the threat out of a river. (2 700 words & fig.)

1932 385 .13 (.73)
ailway Age, No. 9, August 27, p. 284.
DUNN (S. O.). — Taxation and the railroads. (2 300 ords.)

1932 385 .21 (.73)
ailway Age, No. 9, August 27, p. 287.
Dispose of inland waterways. (3 600 words.)

1932 621 .132.8 (.73) & 621 .134.3 (.73)
ailway Age, No. 9, August 27, p, 290.
Development of the multi-pressure locomotive. (3 300 ords & fig.)

1932 656 .2 (.73)
ailway Age, No. 9, August 27, p. 296.
Railway express agency begins highway freight serv-e. (2 400 words, tables & fig.)

1932 621 .139 (.73), 625 .18 (.73) & 625 .27 (.73)
ailway Age, No. 10, September 3, p. 318.
Supply work well organised on Pere Marquette. 1 000 words & fig.)

1932 625 .122 (.73)
ailway Age, No. 10, September 3, p. 323.
Erie Railroad solves severe erosion problem in lateral reams. (3 800 words & fig.)

1932 621 .132.8 (.73) & 621 .134.3 (.73)
ailway Age, No. 10, September 3, p. 329.
Development of the multi-pressure locomotive. (2 700 ords & fig.)

1932 656 .256.3 (.73)
ailway Age, No. 10, September 3, p. 336.
Pennsylvania expedites trains with signaling on 132-ile line. (2 000 words & fig.)

ailway Engineering and Maintenance. (Chicago.)

1932 625 .171 (.73) & 656 .259 (.73)
ailway Engineering and Maintenance, August, p. 492.
Slide detector fences prove practical and effective. 2 800 words & fig.)

1932 621 .392 (.73) & 624 .32 (.73)
ailway Engineering and Maintenance, August, p. 495.
Truss spans strengthened by electric welding. (1 200 ords & fig.)

1932 625 .122 (.73)
Railway Engineering and Maintenance, August, p. 498.
NAGLE (P.). — Embankments constructed by hydraulic methods. (2 600 words & fig.)

1932 621 .132.3 (.41)
Railway Engineering and Maintenance, August, p. 501.
Shutting off bad water in a deep well. (1 200 words.)

1932 721 .6
Railway Engineering and Maintenance, August, p. 502.
What kind of a platform ? (2 600 words & fig.)

1932 614 .8 (.73)
Railway Engineering and Maintenance, August, p. 506.
DORSEY (J. C.). — Stressing the individual in safety work. (1 800 words.)

1932 625 .113
Railway Engineering and Maintenance, August, p. 507.
WALKER (Th.). — A graphic method in string lining curves. (1 000 words & table).

1932 625 .122
Railway Engineering and Maintenance, September, p. 544.
Rock dams remove menace of hillside streams. (1 500 words & fig.)

1932 625 .142.2
Railway Engineering and Maintenance, September, p. 547.
BURTON (W. J.). — What is a softwood tie ? (1 600 words, 1 table & fig.)

1932 625 .13 (.73)
Railway Engineering and Maintenance, September, p. 549.
Modernizing branch line bridges. (2 300 words & fig.)

1932 621 .133.7 (.73)
Railway Engineering and Maintenance, September, p. 552.
Track pans without the usual problems. (2 700 words & fig.)

1932 625 .123 (.73)
Railway Engineering and Maintenance, September, p. 557.
Why some roadbeds hold water and how to get rid of it. (2 700 words & fig.)

Railway Gazette. (London.)

1932 621 .43 & 656 .1
Railway Gazette, No. 7, August 12, p. 188.
A new road-railcar. (250 words & fig.)

1932 621 .132.3 (.41)
Railway Gazette, No. 7, August 12, p. 189.
Locomotive performance on the Great Northern Railway (Ireland). (2 500 words, 1 table & fig.)

1932 **621 .138.2 (.42)**
Railway Gazette, No. 7, August 12, p. 192.
New **coaling plant** at Hull, London & North Eastern Ry. (800 words & fig.)

1932 **621 .335 & 621 .43**
Railway Gazette, No. 7, August 12, p. 193.
Diesel-electric characteristics. (750 words & fig.)

1932 **621 .335 (.489)**
Railway Gazette, No. 7, August 12, p. 194.
Diesel-electric locomotives for Denmark. (1 500 words & fig.)

1932 **621 .43**
Railway Gazette, No. 7, August 12, p. 197.
STUART MIALL. — **Transmissions for Diesel locomotives and railcars.** (1 500 words & fig.)

1932 **656 .1 (06 (.42) & 656 .2 (06 (.42)**
Railway Gazette, No. 8, August 19, p. 217.
Conference on road and rail goods transport. (4 800 words.)

1932 **621 .33 (06 (8)**
Railway Gazette, No. 8, August 19, p. 220.
Electrification of steam railways in all countries. (350 words & tables).

1932 **621 .91**
Railway Gazette, No. 8, August 19, p. 225.
Machine tools for railway workshops. (1 400 words & fig.)

1932 **624 .8 (.82)**
Railway Gazette, No. 8, August 19, p. 227.
The Rio Negro **bridge.** (1 200 words & fig.)

1932 **621 .132.3 (.438)**
Railway Gazette, No. 8, August 19, p. 229.
New Polish **(4-8-2) express locomotive.** (400 words & fig.)

1932 **656 .251 (.43)**
Railway Gazette, No. 9, August 26, p. 245.
RELEAUX (E. W.). — **Signal reform in Germany.** (3 200 words & fig.)

1932 **656 .254**
Railway Gazette, No. 9, August 26, p. 249.
The **Kofler automatic train stop.** (1 200 words & fig.)

1932 **656 .261 (.42)**
Railway Gazette, No. 9, August 26, p. 252.
Modern **railway cartage activities.** (2 500 words & fig.)

1932 **621 .132.5 (.61)**
Railway Gazette, No. 9, August 26, p. 257.
New **2-8-2 type locomotives** for the Moroccan Rys. (800 words & fig.)

1932 **385 .11 (.42)**
Railway Gazette, No. 10, September 2, p. 273.
Railway **operating efficiency.** (2 200 words & tables.)

1932 **625 .233 (.4**
Railway Gazette, No. 10, September 2, p. 276.
Railway coach with sunshine roof. (800 words).

1932 **656 .1 & 656 .2(**
Railway Gazette, No. 10, September 2, p. 278.
Transport of building materials by rail and roa
(1 700 words & fig.)

1932 **621 .392 & 6**
Railway Gazette, No. 10, September 2, p. 283.
Strengthening of weak iron and steel bridges by ele
tric welding. (1 800 words & fig.)

1932 **621 .33 (.42)**
Railway Gazette, No. 10, September 2, p. 301.
The Barking and Upminster **electrification,** Tilbur
section, London Midland & Scottish Ry. (2 600 wor
& fig.)

1932 **656 .211.5 (.42**
Railway Gazette, No. 11, September 9, p. 307.
Sudbury **town station.** (1 000 words & fig.)

1932 **621 .4**
Railway Gazette, No. 11, September 9, p. 311.
STUART MIALL. — **Transmissions for Diesel loco**
motives and railcars. (1 400 words & fig.)

1932 **621 .43 (.43**
Railway Gazette, No. 11, September 9, p. 313.
Narrow-gauge **geared Diesel locomotive.** (1 300 word
& fig.)

1932 **621 .43 (.43**
Railway Gazette, No. 11, September 9, p. 315.
HEINZE (E. P. A.). — **New 410 B. H. P. ligh**
weight Diesel engine. (1 200 words & fig.)

Railway Magazine. (London.)

1932 **656 .222.1 (.4**
Railway Magazine, September, p. 199.
Continental summer train services, 1932. (3 600 word
& tables).

Railway Mechanical Engineer. (New York.)

1932 **625 .234 (.73)**
Railway Mechanical Engineer, August, p. 305.
Illinois Central **de luxe cars are air cooled.** (2 30
words & fig.)

1932 **621 .335 (.73) & 621 .43 (.73)**
Railway Mechanical Engineer, August, p. 309.
Santa Fe **installs 900 h. p. articulated rail car.** (2 00
words, 4 tables & fig.)

1932 **621 .43 (.73)**
Railway Mechanical Engineer, August, p. 316.
Gasoline locomotives for light switching. (400 words,
1 table & fig.)

1932 625 .246 (.73)
Railway Mechanical Engineer, August, p. 317.
All-steel box-car design presented to Mechanical Division, American Railway Association. (3 500 words, tables & fig.)

1932 621 .138
Railway Mechanical Engineer, September, p. 345.
COOK (T. R.). — **How age affects locomotive maintenance costs.** (2 400 words, 2 tables & fig.)

1932 625 .244 (.73)
Railway Mechanical Engineer, September, p. 349.
TRACY (H. E.). — Pacific Fruit Express Company builds seventy-ton refrigerator cars. (2 700 words & fig.)

1932 625 .214 (.73)
Railway Mechanical Engineer, September, p. 353.
BUCKWALTER (T. V.). — **Roller bearings on railroad equipment.**

Railway Signaling. (Chicago.)

1932 656 .257 (.73)
Railway Signaling, August, p. 231.
Baltimore & Ohio Chicago Terminal Railroad **stops theft of coal with four automatic plants** at Chicago Heights, Ill. (1 500 words & fig.)

1932 625 .162 (.82) & 656 .254 (.82)
Railway Signaling, August, p. 234.
WEBSTER (J. A.). — An experiment in **highway-crossing signal control circuits** in the Argentine. (1 200 words & fig.)

1932 656 .257 (.73)
Railway Signaling, August, p. 235.
Boston & Maine **interlocking** at Boston. (5 000 words & fig.)

1932 625 .171 (.73) & 656 .259 (.73)
Railway Signaling, August, p. 243.
Slide-detector fences on the Norfolk & Western. (2 400 words & fig.)

1932 656 .257 (.73)
Railway Signaling, August, p. 247.
The relation of maintenance to performance in an electric interlocking plant. (2 000 words & fig.)

1932 625 .258 (.73)
Railway Signaling, September, p. 263.
ZANE (W. F.). — **Car retarders** on the Burlington. (3 000 words & fig.)

1932 656 .254 (.73)
Railway Signaling, September, p. 271.
Interstate Commerce Commission permits **cab signals in lieu of automatic train control** on the Union Pacific. (2 300 words.)

The Locomotive. (London.)

1932 621 .132.5 (.42)
The Locomotive, August 15, p. 267.
Three-cylinder 2-8-0 mineral engine, London & North Eastern Ry. (300 words & fig.)

1932 621 .132.8 (.44)
The Locomotive, August 15, p. 268.
Beyer-Garatt express locomotive P. L. M. Ry. (Algeria). (2 500 words & fig.)

1932 621 .131.2
The Locomotive, August 15, p. 277.
PHILLIPSON (E. A.). — **Steam locomotive design : data and formulæ.** (1 800 words.)

1932 621 .43 (.41)
The Locomotive, August 15, p. 280.
Diesel rail-cars, Great Northern Ry. (Ireland). (750 words & fig.)

1932 621 .133.2 (.44) & 621 .392 (.44)
The Locomotive, August 15, p. 282.
Welded copper firebox, Eastern Ry. of France. (900 words & fig.)

1932 625 .246 (.42)
The Locomotive, August 15, p. 283.
40-ton bogie well trollies, London Midland & Scottish Ry. (250 words & fig.)

1932 621 .331 (.42) & 621 .43 (.42)
The Locomotive, August 15, p. 287.
Oil-electric locomotive for the Ford Motor Co.'s Works. (1 200 words & fig.)

1932 621 .132.3 (.45)
The Locomotive, August 15, p. 300.
« Mikado » and « Pacific » type locomotives, Italian State Rys. (2 600 words & fig.)

Transit Journal. (New York.)

1932 388 (.73) & 656 .1 (.73)
Transit Journal, August, p. 342.
Adjusting **bus service to traffic demands.** — **Advantages and disadvantages of buses of various sizes.** (1 500 words, tables & fig.)

1932 621 .333 (.73)
Transit Journal, August, p. 345.
KEARNS (E. E.). — **Increasing car speeds with shunted motor fields.** (500 words & fig.)

1932 625 .255 (.73)
Transit Journal, August, p. 346.
WARD (F. T.). — **Brake tests point way to faster schedules.** (3 000 words & fig.)

1932 614 .8 (.73)
Transit Journal, August, p. 352.
DANA (E.). — **Accidents reduced by research methods.** (1 400 words & fig.)

1932 **625 .4 (.73)**
Transit Journal, September, p. 371.
New York to open its latest **subway.** (2 000 words & fig.)

1932 **625 .4 (.73) & 656 .256.3 (.73)**
Transit Journal, September, p. 374.
Automatic signals and interlocking give protection to subway. (1 600 words & fig.)

1932 **625 .26 (.73)**
Transit Journal, September, p. 377.
Modern **shops serve independent subway system.** (1 000 words & fig.)

1932 **621 .31 (.73)**
Transit Journal, September, p. 380.
Central station **power used for new subway.** (1 800 words & fig.)

In Spanish.

Ferrocarriles y Tranvias. (Madrid.)

1932 **385 (.460)**
Ferrocarriles y Tranvias, agosto, p. 245.
REYES (F.). — I. — La fácil **transformación** de la red ibérica. (3 800 palabras & fig.)

1932 **656 .254**
Ferrocarriles y Tranvias, agosto, p. 252.
MAURIÑO (F. G.). — Instrucciones sobre **passos a nivel** de los ferrocarriles suecos. (2 600 palabras & fig.)

Gaceta de los Caminos de hierro. (Madrid.)

1932 **656 (.73)**
Gaceta de los Caminos de hierro, nº 3688, 15 de agosto, p. 181; nº 3689, 1º de septiembre, p. 193.
La **explotación** de los ferrocarriles en los Estados Unidos y la competencia y coordinación con otras medios de transporte. (4 800 palabras.) (Continuará.)

Ingenieria y Construcción. (Madrid.)

1932 **656 .1 & 656 .2**
Ingenieria y Construcción, septiembre, p. 473.
Medios que en los ferrocarriles se proyectan o emplean para contrarrestar la **competencia** o encauzar el tráfico. (5 200 palabras.)

Revista de Obras Públicas. (Madrid.)

1932 **656**
Revista de Obras Publicas, nº 17, 1º de septiembre, p. 409.
ROMERA (J.). — **Problemas ferroviarios.** (2 800 palabras & fig.)

1932
Revista de Obras Publicas, nº 18, 15 de septiem[bre], p. 426.
ROMERA (J.). — **Problemas ferroviarios.** (3 000 [pa]labras & fig.)

1932 **656 .1 & 656**
Revista de Obras Publicas, nº 18, 15 de septiem[bre], p. 434.
ONTIVEROS (F. J.). — La **carretera y el ferroca[rril]** (2 400 palabras.)

In Italian.

Annali dei lavori pubblici. (Roma.)

1932 **624 .63 (.4**
Annali dei lavori pubblici, maggio, p. 376.
ORIOLI (V.). — Un nuovo **ponte** sull isarco a pr[esso] isarco. (5 700 parole & 7 quadri.)

1932 **62. (**
Annali dei lavori pubblici, maggio, p. 416.
TAGLIACOZZO (C.). — Contributo al calcolo de[gli] **involucri elastici.** Tensioni in un involucro resisten[te] alla flessione avente per superficie media un para[bo]loide di rotazione. (4 100 parole & fig.)

L'Ingegnere. (Roma.)

1932 **669**
L'Ingegnere, agosto, p. 545.
TRANSINI (I.). — La **nitrurazione dell' acciai[o]** (6 500 parole & fig.)

1932 **62. (0**
L'Ingegnere, agosto, p. 572.
ORABONA (F.). — Contributo al calcolo delle **stru[t]ture cilindriche.** (4 400 parole & fig.)

1932 **624. (0**
L'Ingegnere, agosto, p. 583.
BONVICINI (D.). — Formule per il calcolo d[el] portale incastrato tipo « cattedrale ». (3 500 parole [&] fig.)

Rivista tecnica delle ferrovie italiane. (Roma.)

1932 **625 .244 (.45**
Revista tecnica delle ferrovie italiane, 15 agosto, p. 107
PERFETTI (A.). — La determinazione sperimental[e] della trasmissione del calore attraverso i carri isoter[mici] e refrigeranti in esercizio corrente e del loro equi[va]lente in acqua. (4 300 parole & fig.)

1932 **621 .3**
Revista tecnica delle ferrovie italiane, 15 agosto, p. 119
CUZZER (O.). — Metodo differenziale assoluto per l[a] verifica dei **riduttori di corrente.** (1 600 parole & fig.)

In Dutch.

De Ingenieur. (Den Haag.)

1932 **624 .2 (01**

e Ingenieur, n° 34, 19 Augustus, p. 166.

RANX (C.). — **Berekening van meervoudig statisch epaalde staafverbindingen** met stijve knooppunten r John E. Goldberg. (2 400 woorden & fig.)

1932 **621 .132.8 (.492)**

Ingenieur, n° 35, 26 Augustus, p. 71.

RUNNER. — De eerste **Garrat-locomotief** in Neland. (4 300 woorden & fig.)

1932 **621 .33**

Ingenieur, n° 35, 26 Augustus, p. 78.

ERSCHOOR (E.). — Spitsverkeer en **electrificatie.** 0 woorden & 1 tabel.)

Spoor- en Tramwegen. (Utrecht.)

1932 **621 .43 (.73)**

oor- en Tramwegen, n° 5, 30 Augustus, p. 109; n° 6, 13 September, p. 148.

ORPORAAL (E.). — Amerikaansche exploitatie van **torrijtuigen.** (5 000 woorden & fig.) (Slot volgt.)

1932 **656 .256.3 (.492)**

Spoor- en Tramwegen, n° 6, 13 September, p. 137.

VERSTEGEN (J.). — **Automatisch blokstelsel** op de Nederlandsche Spoorwegen. (2 800 woorden & fig.) (Wordt vervolgd.)

In Portuguese.

Gazeta dos caminhos de ferro. (Lisboa.)

1932 **385. (09 (.68)**

Gazeta dos caminhos de ferro, n° 1072, 16 de agosto, p. 377.

GALVAO (L.). — O **caminho de ferro** da Rhodesia a Walfish-Bay. (1 000 palavras.)

Revista das Estradas de ferro. (Rio de Janeiro.)

1932 **385 (.81)**

Revista das Estradas de ferro, n° 169, 30 de julho, p. 259.

VIEIRA (F.). — Suggestões sobre a systematisação de um **plano geral** para a viação ferrea brasileira. (3 300 palavras.)

13.459. — M. Weissenbruch Co., Ltd., Printer to the King, 49, rue du Poinçon, Bruxelles.

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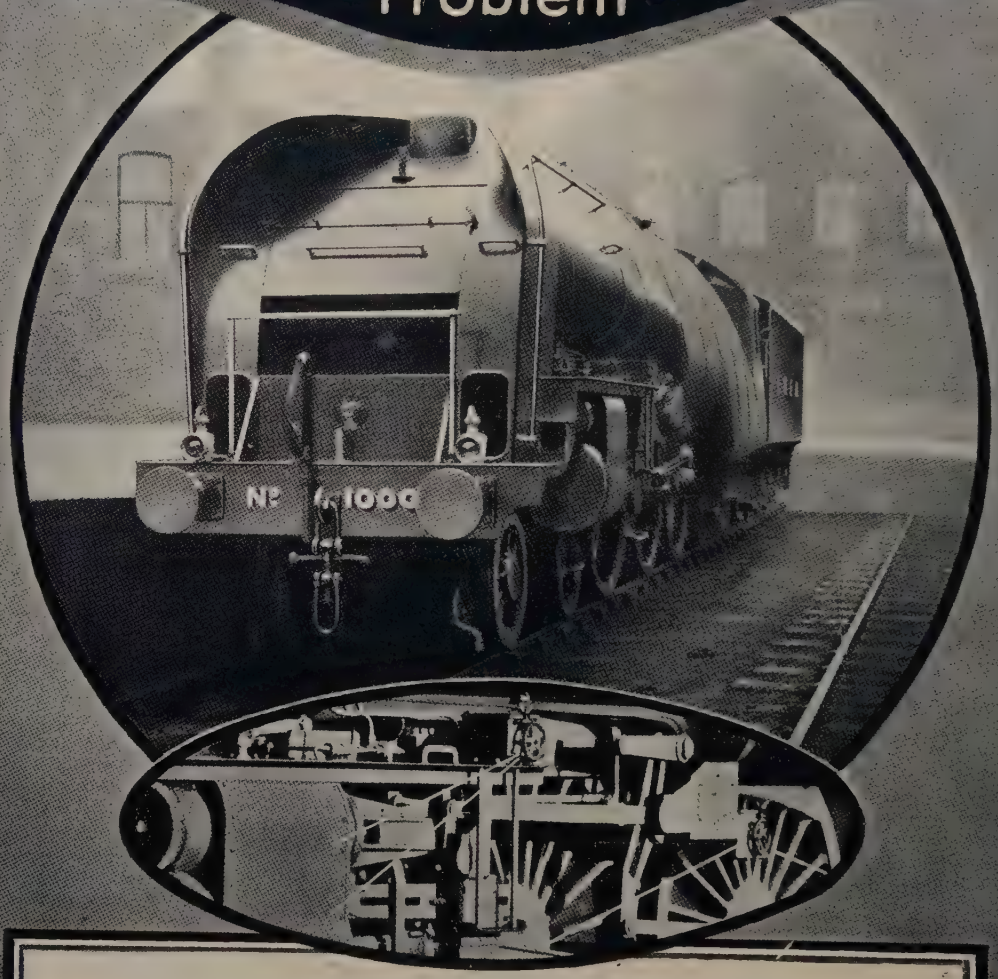

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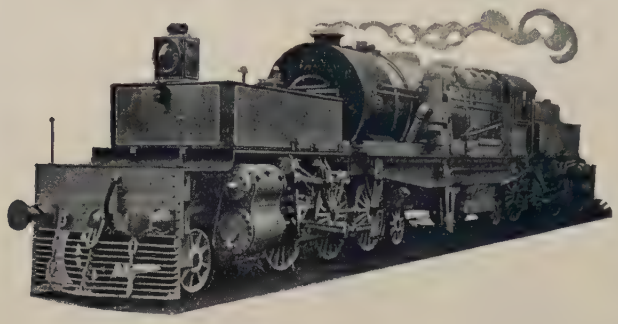

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This 40-ton six-coupled Shunter has been in service for 13 weeks, 24 hours a day, six days a week, in the London & North Eastern Railway Yards. It has handled loads up to 816 tons up a gradient of 1 in 150.
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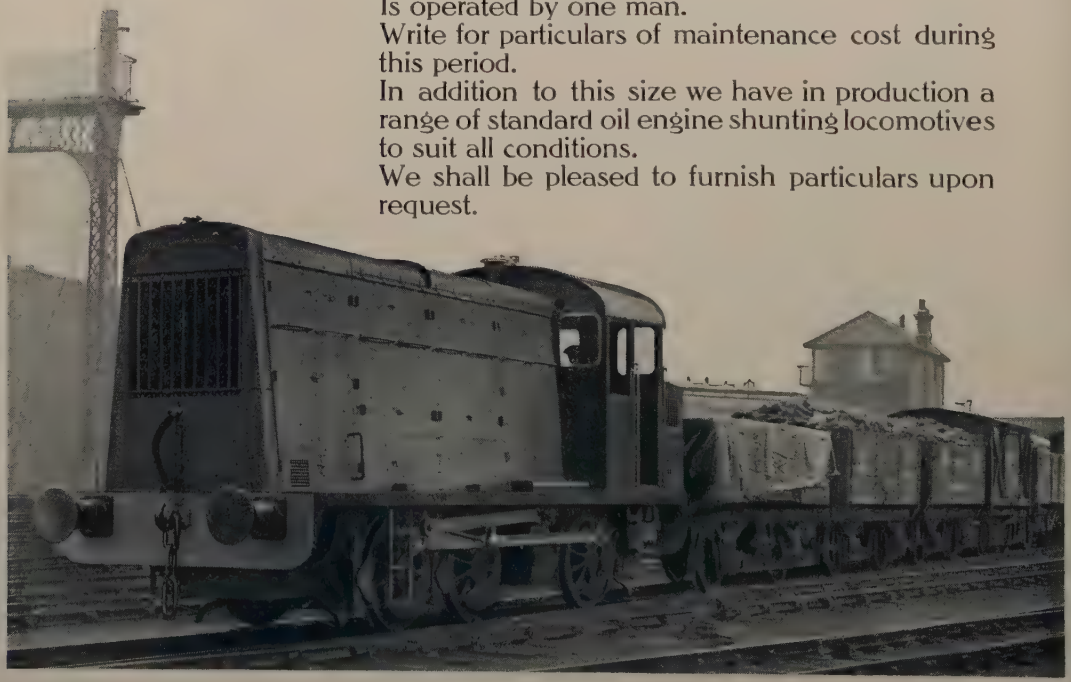

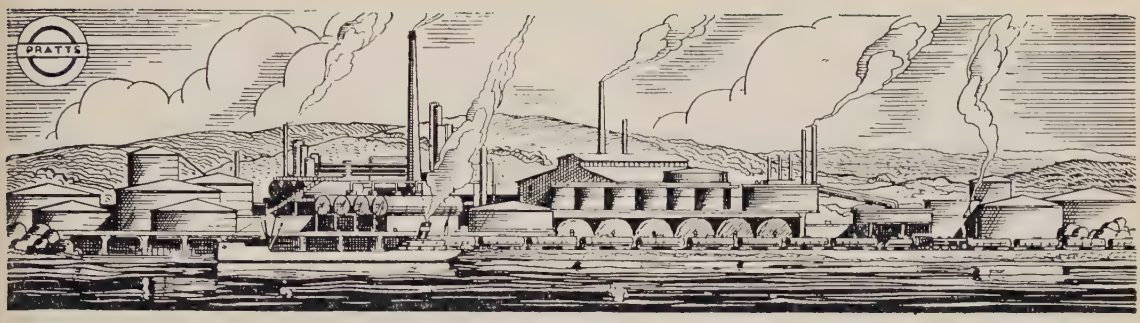

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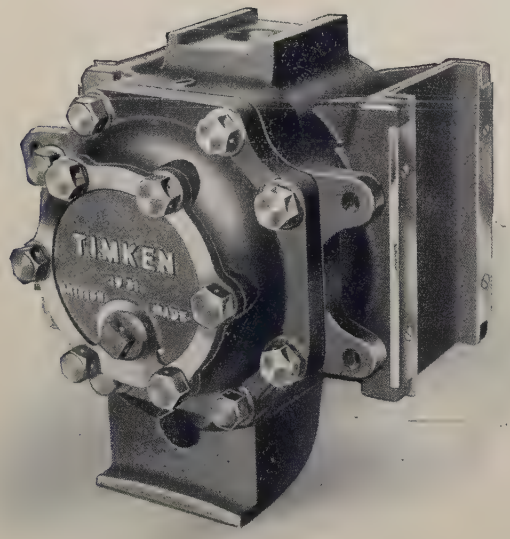

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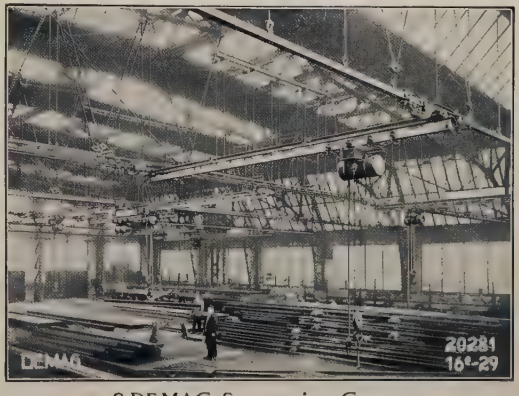
Two 50 ton locomotive transporting cranes.

Sunk traverser with five platform supports.

Five ton gantry crane in the raw material storeyard of a railway workshop.



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IMMEDIATE RESULTS

On September 5th ordinary trains on the **Craiova-Calafat** section of the Roumanian railways were replaced by **"Sentinel-Cammell" Railcars.**

During the previous month the average number of passengers carried daily by the ordinary trains was 268, or an average of 45 per train.

From September 5th to 14th the railcars carried a daily average of 418 passengers, or 70 per trip.

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Fares have been reduced by 50 per cent., but the figures show that the **revenue has actually been increased,** while the running expenses are stated to have been reduced in the proportion of 7 : 2.

Write for full particulars to :

The "Sentinel" Waggon Works Ltd.

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Branch of
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CUMBERLAND, ENGLAND

THE UNITED STEEL COMPANIES LTD

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The present range of Workington Steel Sleepers includes a variety of carefully developed types covering every need, for both bull-head and flat-bottomed rails. (Write for publication W.103).

Each of the Big Four English railway companies is now using Workington Steel Sleepers, as many railway systems in other parts of the world have done for years past.

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One of the Beardmore Diesel-Electric Rail Cars in Spain

BEARDMORE ENGINED UNITS HAVE RUN
OVER **4,000,000** MILES
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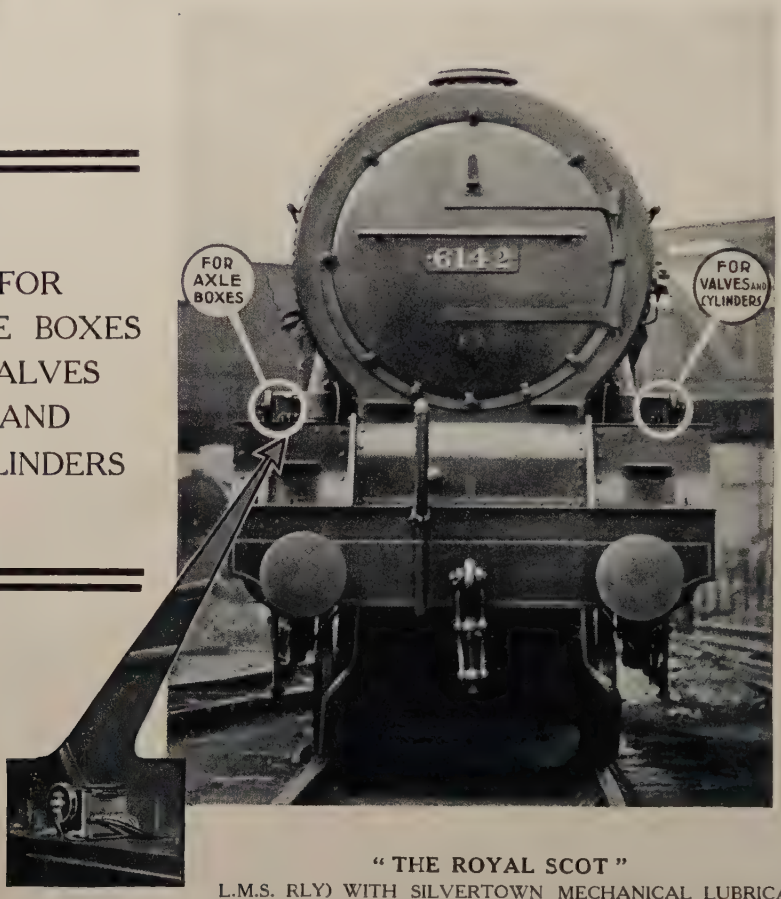

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Research work carried out by the British Non-Ferrous Assn. has proved that failure of fire box stays due to heads wearing is almost entirely the result of corrosion caused by leaky stays.

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6 Cyl.
1400 r. p. m.
Weight,
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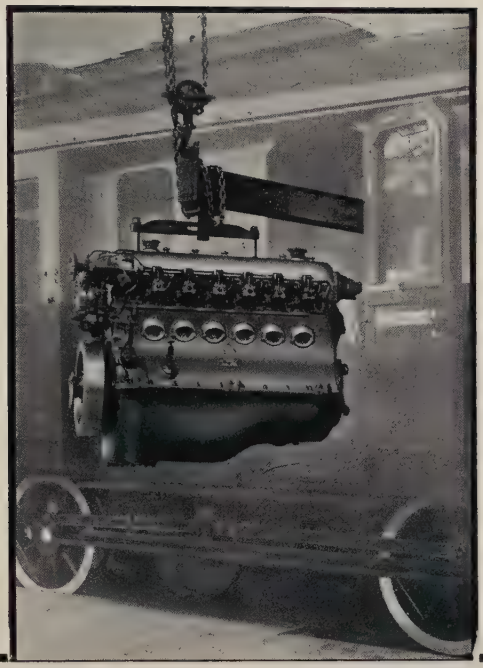

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Room taken up by the 175 H. P. Diesel :

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Stations serving the Paris Main Railway Stations

(Continued.)

16th Arrondisment (continued) **PASSY**

La Muette Quarter :

La Muette	Line	9.
Passy	»	5.
Ranelagh	»	9.

Porte Dauphine Quarter :

Pompe	Line	9.
Porte Dauphine	»	2.

Chaillot Quarter :

Boissière	Line	5.
Kléber	»	5.
Iéna	»	9.
Porte Maillot	»	1.
Trocadéro	»	5 and 9.
Victor-Hugo	»	2.

17th Arrondisment. BATIGNOLLES-MONCEAU.

Ternes Quarter :

Etoile	Lines	1, 2 and 5.
Obligado	»	1.
Porte Champerret	»	3.

Plaine-Monceau Quarter :

Malesherbes	Line	3.
Péreire	»	3.
Villiers	»	2 and 3.
Wagram	»	3.

Batignolles Quarter :

Place Clichy	Lines	2 and 13.
Rome	»	2.

Epinettes Quarter :

Brochant	Line	13bis.
La Fourche	»	13 and 13bis.
Porte de Clichy	»	13bis.
Porte de Saint-Ouen	»	13.

18th Arrondisment. BUTTE-MONTMARTRE.

Grandes Carrières Quarter :

Blanche	Line	2.
Lamarck - Caulaincourt	»	12.
Marcadet-Balagny	»	13.

Clignancourt Quarter :

Abbesses	Line	12.
Anvers	»	2.
Château-Rouge	»	4.
Porte de Clignancourt	»	4.
Jules-Joffrin	»	12.
Simplon	»	4.

Goutte d'Or Quarter :

Barbès-Rochechouart	Lines	2 and 4
Porte de la Chapelle-Saint-Denis	»	12.
Marcadet-Poissonniers	»	4 and 12.

Chapelle Quarter :

Torcy	Line	12.

19th Arrondisment. BUTTES-CHAUMONT.

La Villette Quarter :

Aubervilliers	Lines	2 and 7bis.
Crimée	»	7bis.
Jean-Jaurès	»	2 and 7.
Riquet	»	7bis.
Boul. de la Villette	»	2 and 7bis.

Pont de Flandre Quarter :

Pont de Flandre	Line	7bis.
Porte de la Villette	»	7bis.

Amérique Quarter :

Place du Danube	Line	7.
Place des Fêtes	»	7.
Pré Saint-Gervais	»	7.

Combat Quarter :

Bolivar	Line	7.
Botzaris	»	7.
Buttes-Chaumont	»	7.
Combat	»	2.

20th Arrondisment. MENILMONTANT.

Belleville Quarter :

Belleville	Line	2.

St-Fargeau Quarter :

Porte des Lilas	Line	3.
Saint-Fargeau	»	3.

Père Lachaise Quarter :

Père-Lachaise	Lines	2 and 3.
Gambetta	»	3.
Martin-Nadaud	»	3.
Pelleport	»	3.

METROPOLITAN STATIONS SERVING THE MAIN RAILWAYS STATIONS OF PARIS

Gare de l'Est	Lines	4, 6, 7 and 7bis.
Invalides	»	8 and 10.
Gare de Lyon	»	1.
Montparnasse	»	4 and 12.
Gare du Nord	»	4 and 6.
Gare d'Orléans-Austerlitz	»	6.
Saint-Lazare	»	3, 12, 13 and 13bis.
Bastille (Gare de Vincennes)	»	1, 6 and 8.
Solférino (near the Orsay Station)	»	12.

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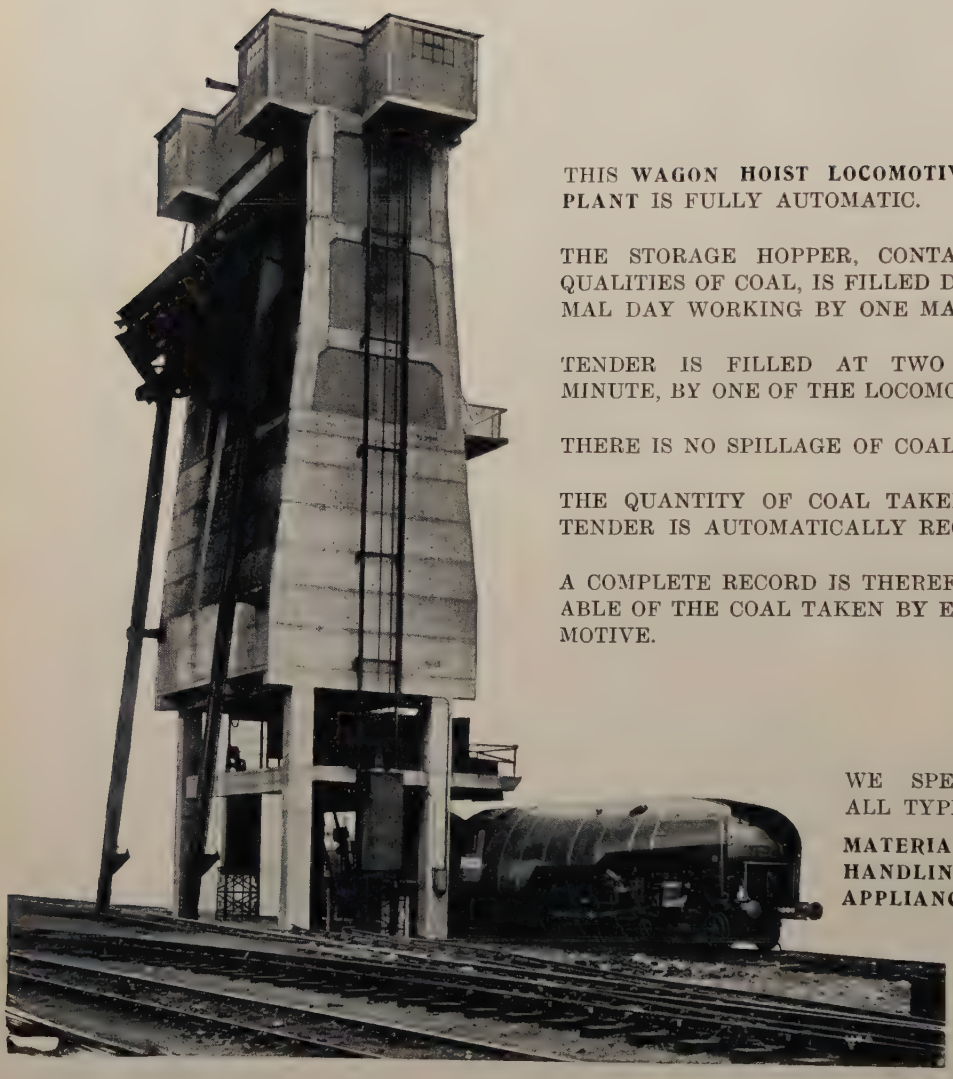

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PARIS : 14, rue du 4 Septembre.
ANTWERPEN : Station Antwerpen (Dokken en Stapelplaats).
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with
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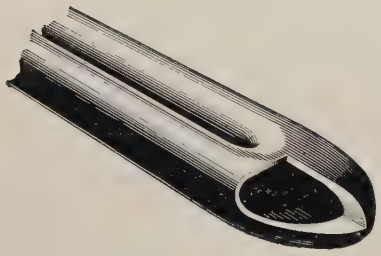
FIRETUBE SUPERHEATERS

Integrally Forged Return Bends



Long experience has demonstrated conclusively the superiority of the design and construction of Elesco (Schmidt) superheaters, and their reliability and efficiency in operation. This is due largely to exclusive use of return bends, integrally forged with the tubing which form the loops of the superheater elements or units. These bends are machine-forged without utilizing either additional material or a flux.

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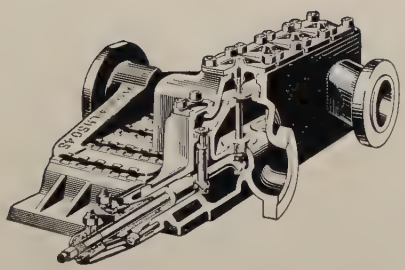

Headers for Elesco superheaters are made of the very best gray iron and are designed to meet internal stresses to which they are subjected through constant vibration and variations in temperature.

A recent development of the finger type header has the multiple-valve throttle or regulator integral with the casting, as illustrated. Without complicating the smokebox arrangement, this throttle or regulator offers many advantages over other designs. the steam is controlled between the superheater and the cylinders, and by the consecutive opening and closing of a series of small valves the steam supply to the cylinders is finely graduated. Steam at full boiler pressure constantly circulates through the units, protecting them from overheating. The multiple-valve throttle-header is being adopted rapidly for every class of locomotive service.

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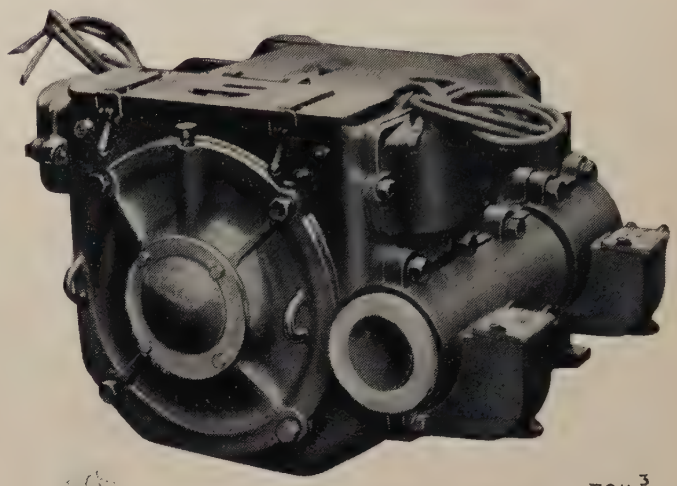

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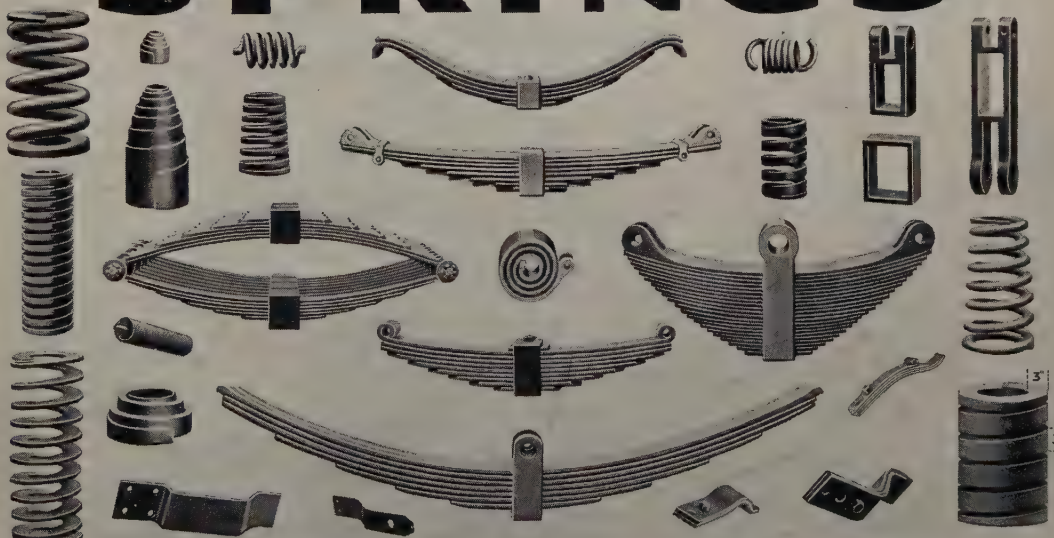

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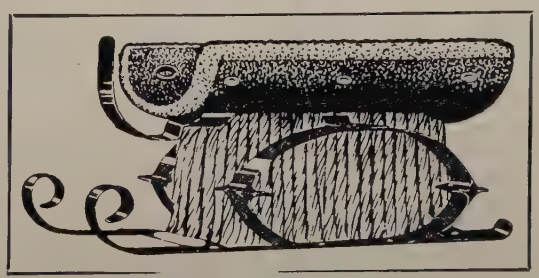

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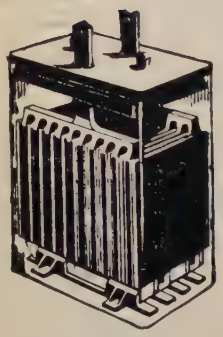

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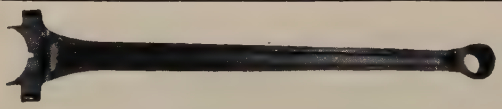
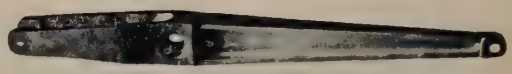


Locomotive connecting rod in " R.R. 56,, alloy



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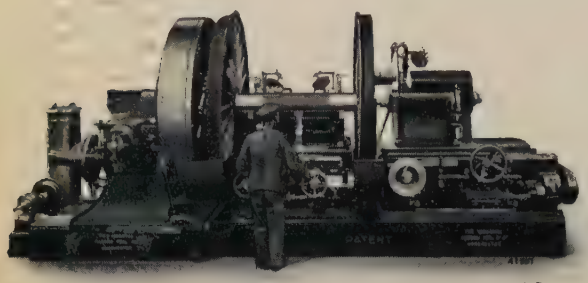

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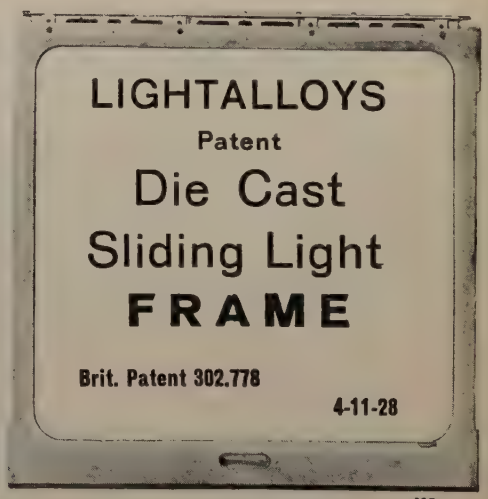

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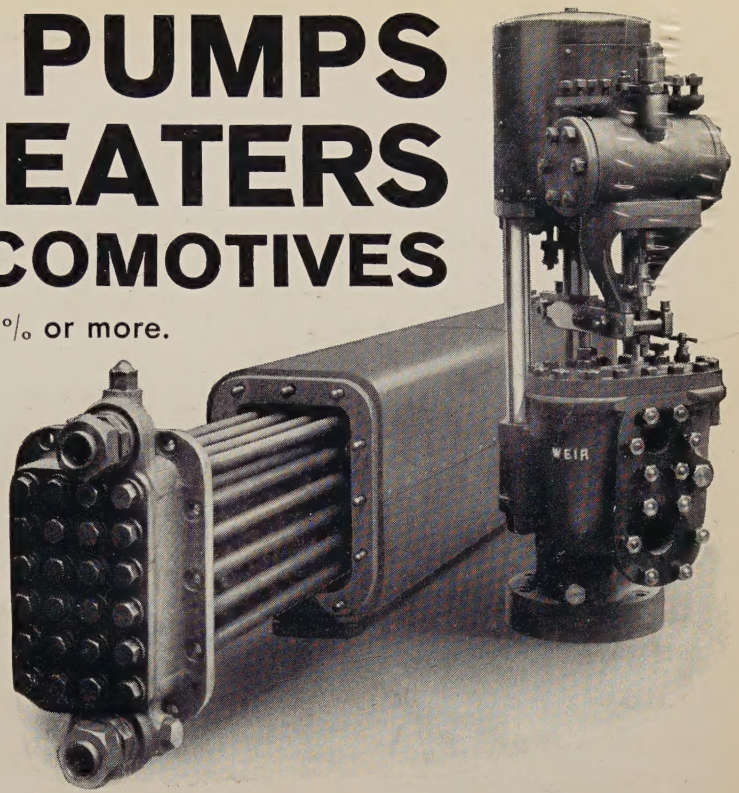

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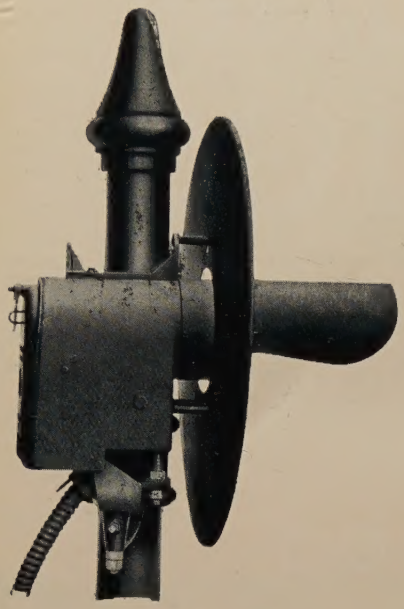

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